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ASSESSMENT OF ADVANCED BUS SUSPENSION TECHNOLOGY

VOLUME 1: MAIN REPORT

BY

J.H. STANNARD, B.T. WHITTEN
J.E. TREGENZA

UTDC INC.

PREPARED FOR: TRANSPORT CANADA, TRANSPORTATION DEVELOPMENT CENTRE

DSS CONTRACT SERIAL NO.: 1SD85-00205

UTDC INC. REPORT NO.: TR 449-625-001

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Un sommaire en français de ce document est inclus avant la table des matières.

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17. Abstract

The objective of this study was to quantify the benefits of advanced suspension systems when compared to existing North American suspension systems on highway buses. Existing computer simulation modules were modified where necessary and integrated into an overall program capable of simulating vehicle ride and handling in considerable detail at the component level. The simulation program was verified by a test program on an existing MCI 102-A3 intercity bus from which a variety of test results for steady state and transient handling and ride data were accumulated. The successfully verified model was used to assess a number of advanced suspension systems recommended for investigation as a result of a state-of-the-art survey carried out during the study.

Increased roll stiffness and damping is recommended for near-term development. It was concluded that the level of improvement possible with an independent front suspension and A-frame rear suspension was desirable but would require a longer term development as part of an advanced bus program.

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17. Résumé

La recherche dont il est rendu compte ici avait pour objectif de chiffrer les avantages que peuvent apporter des systèmes avancés de suspension pour autocars, par rapport aux systèmes existants en Amérique du Nord. Des modules de simulation informatique dûment adaptés ont été utilisés pour représenter à un niveau de détail très poussé la directibilité et la tenue de route des véhicules équipés des suspensions étudiées. Les résultats théoriques obtenus ont été validés par des essais routiers effectués avec un autocar MCI 102-A3 et visant à connaître les caractéristiques particulières de sa suspension en régime permanent et transitoire ainsi que sa tenue de route. Ainsi validé, le modèle de simulation a servi à caractériser un certain nombre de systèmes avancés de suspension qu'un tour d'horizon avait permis de recenser.

A court terme, il est préconisé d'accroître les raideurs en roulis et l'amortissement. La recherche a montré que les améliorations possibles grâce à une suspension avant à roues indépendantes et à une suspension arrière à cadre triangulaire sont souhaitables, mais que des travaux plus poussés sont nécessaires dans le cadre d'un programme d'étude d'un autocar de type avancé.

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SUMMARY

The work described in this report consisted of an in-depth analysis of advanced suspension systems for highway buses. This study ensued in part from work undertaken by UTDC Inc. (UTDCI) for the Transportation Development Centre (TDC) to complete the conceptual design work required to manufacture a Canadian, advanced technology double-deck intercity coach. The project takes advantage of previous work in this field by UTDCI and other contractors to TDC. The work complements work done by the National Research Council (NRC) in truck suspension/parametric measurements and by Concordia University in the seat ride vibration analysis.

The overall objective was "to contribute to more efficient and attractive highway bus transportation and thereby improve the health of the carrier industry, reduce total transportation costs and encourage greater public use of this economical and energy efficient mode".

This project addresses a crucial subcomponent area to make advances in the state-of-the-art with respect to bus suspension system design. The specific objectives of this project were to:

- quantify the benefits of advanced suspension systems compared to existing North American suspension systems on highway buses.
- improve and generalize the suspension/vehicle interface model as a design tool;
- verify the model by means of road vehicle testing using a typical modern conventional highway bus; and
- make quantitative predictions of the benefits of applying advanced suspension system concepts, including active suspension concepts, to both existing and new design highway buses.

The project approach to the overall objective of improving highway bus transportation was to address the design modelling aspect of the bus suspension. A design tool was developed in which additional components and restructured suspension configurations can be easily examined to predict, with a high level of confidence, component-level design improvements.

The first step of this process was to study state-of-the-art technology in highway bus suspension, particularly in North America and in Europe. Included in the study were current methods for design and analysis, and a review of other heavy vehicle suspensions for new technical ideas and adaptation to highway buses.

The review resulted in the selection of a short list of suspension technology improvements that would be analysed using the computer model; they were:

- independent front suspension
- modified rear suspension with "H-Frame" layout to increase roll stiffness
- modified rear suspension with "A-frame" layout to increase roll stiffness
- adaptive suspension techniques, for example adaptive shock absorber control.

It was concluded during a survey of active suspension techniques that in the near to medium term, fully active suspension was not a viable solution for an intercity bus, largely on grounds of operational reliability, maintenance requirements and high initial cost.

Operational needs and performance requirements, based on a variety of sources, were established for the suspension system of a highway bus, and general targets set as guidelines for advanced designs that would improve or retain economic viability for the operator.

Specifications for a general computer model and analysis of highway bus suspension performance were then formulated, ensuring that the necessary performance criteria be predicted and that sufficient flexibility be incorporated for use as a design tool. Based on these specifications and on the methods of analysis currently available, the most appropriate modelling techniques were selected, modified and adapted to the development of an advanced computer suspension model for any highway bus.

The result of the investigations was a short list of candidate simulation packages. Since it was determined that some software already existed, the option of developing a totally new package specific to buses and with the governing equations derived by hand was eliminated. The remaining options were:

- A set of dynamic simulation programs developed by UMTRI for heavy truck/trailer configurations. These simulations are widely used and validated. They are available in the public domain and versions are running at NRC and at Concordia. The vehicle models are of the lumped parameter type wherein suspension elements are considered in groups and have their properties lumped together as a single element. These simulations are meant for handling performance only, ride quality output not being predicted. The input of suspension properties is tailored to existing vehicles where large-scale static tests can determine the overall behaviour of an installed suspension. It is not possible to simulate complex suspension systems in a detailed fashion using the UMTRI programs.

- The use of one of the programs now available for automatic derivation and solution of equations governing three-dimensional, non-linear, multi-body dynamic systems. Three such packages were considered, the first being the ADAMS package developed by MDI. This package is gaining favour in the United States for its abilities to handle simulations of a complexity and level of detail which could not be considered if equations were derived by hand. The package has a high purchase price, roughly $40,000 US, and requires quite a high level of training in order to operate it efficiently. ADAMS has been used in handling performance predictions of highway vehicles with good results.

  The other two programs considered were MEDYNA, developed in Germany by DFVLR, and RESTRI, a product of Queen's University. The first of these has a purchase price equivalent to that of ADAMS. It has been used primarily for rail vehicle simulations where flexible guideways were of importance making it the only package which currently has the ability to simulate elastic body modes. It requires large scale computing hardware and a high level of operator training. RESTRI is a similar program which is still in a development state. It has been shown to provide good simulations of simple rail vehicles and other dynamic systems. It is able to run on small computer systems but requires a significant level of effort on the part of users before they feel comfortable using it.

- The use of a package which automatically derives equations of motion which have linear geometry (small angle approximations) but include non-linear component properties. The package A'GEM, developed at Queen's University, is highly interactive and capable of simulating large systems on small computers. The solution of the governing equations, once derived, must be handled by another program specifically designed for the simulation at hand. UTDCI has software capable of solving these equations with relatively minor modifications. The initial cost of the A'GEM program is $1200 for source code and significantly less for a microcomputer package.

Based on the above information and the requirements of the design tool, it was decided that the most promising approach involves the use of the A'GEM package coupled with special-purpose solution routines available at UTDCI. The use of the UMTRI software was ruled out because it could not provide the level of suspension detail required by a designer and could not simulate ride quality of a vehicle. The alternative of using a fully non-linear simulation, probably ADAMS if this were the decision, is attractive for reasons of realism in the simulation but was ruled out on the basis of cost of purchase, level of training required of the user, and on-going costs for maintenance and hardware.

The package which was used for the bus simulation therefore took the form of a series of modules, each of which could easily used by the designer and which would provide a variety of output variables in line with those required. The model included non-linear suspension elements thereby limiting the solution technique to the time-domain. The ride quality results, which are typically quoted in the frequency-domain, were obtained by post-processing the time varying output accelerations as would be done for experimental results. The handling performance results are interpreted in the time-domain and require no post-processing. The software necessary to effect the solution and the post-processing existed at UTDCI and was modified to fit into the design package.

Real suspension performance data was acquired by instrumenting and operating a conventional highway bus over a predetermined course and varying the response characteristics of the bus suspension system in both on- and off-design configurations. The measured variations in performance allowed verification of the suspension model through comparison of computer predictions with actual data.

The test program was carried out using an MCI 102-A3 intercity bus supplied by the manufacturer. It was used for a variety of tests including transient and steady state handling and assessment of the ride in a variety of conditions. Vehicle weight, center of gravity and effect of the tag axle were investigated.

The suspension model was further developed and refined until a high level of confidence (credibility) was achieved in the computer predictions. The model and computer analysis were extended to examine the feasibility and performance expectations of improved suspension system designs, potential modifications to existing designs and of the incorporation of active suspension elements into the designs. Performance benefits were identified for each suspension design, and recommendations made for the development of the most promising suspension system for highway buses in North America.

The conclusions resulting from the work are summarized below:

o current North American buses have rugged and long life suspensions to suit road conditions, but improvements to ride and handling are both desirable and practical.

- Experience in Europe indicates that significant improvement is possible with relatively conventional incremental type developments.
- A fully active suspension is unlikely to be a cost-effective solution in the near- or medium-term for intercity buses, and at the current stage of development there are serious reservations about reliability, maintainability and safety.
- The computer simulation technique developed agrees reasonably well with road test data for the MCI 102-A3 test bus in both on- and off-design conditions, but the range of variation in parameters over which it was tested produced smaller effects than expected.
- There is evidence to suggest that body stiffness will need to be modelled for a more accurate representation of real vehicle behaviour.
- The most significant undercar components with a strong effect on simulation results were the shock absorbers. It is also significant to note that there was a marked and undesirable variation in these components on the test bus.
- For effective simulation the non-linearity of the shock absorber and coulomb-type damping must be modelled.
- Some form of active damping control is a low cost and feasible technique for improving handling in manoeuvering while not causing a deterioration in ride in normal operation.
- Based on the modelling results of advanced suspensions, the most promising suspension layout for an advanced bus comprises independent front suspension, pusher type tag axle, and A-frame style drive axle suspension.
- Large improvements in suspension performance are unlikely without major bus redesign.
- Some worthwhile improvements may still be made by increasing roll stiffness and damping by incorporation of an A-frame or wider based H-frame suspension on existing bus designs.

It is recommended that:

- Future modelling of buses incorporate at least some representation of body stiffness. (The developed program is capable of this.)
- A near-term program should focus on increasing roll stiffness and damping at the rear of the vehicle.
- A longer-term program should focus on independent front suspension with an A-frame drive axle and pusher type tag axle.

SOMMAIRE

La recherche dont il est rendu compte ici concerne l'étude analytique fouillée de systèmes avancés de suspension pour autocars. Elle est le prolongement des travaux faits antérieurement par UTDC Inc., pour le compte du Centre de développement des transports (CDT), dans le cadre d'une étude conceptuelle visant la fabrication au Canada d'un autocar à impériale, de type avancé. Elle s'est inspirée des travaux de l'UTDC dans le domaine considéré ainsi que d'autres effectués pour le CDT. Enfin, elle a été menée parallèlement aux travaux du Conseil national de recherches sur l'analyse paramétrique des systèmes de suspension pour camions, et à ceux de l'Université Concordia concernant la caractérisation de l'aptitude des sièges de véhicules routiers à réduire les vibrations transmises au conducteur.

L'objectif général fixé à cette recherche était "de relever l'économie et l'attrait du transport routier par autocar, d'améliorer la situation financière de l'industrie concernée par la réduction du coût global et d'inciter le public voyageur à privilégier ce mode de transport économique à tous les points de vue".

Quant à l'objectif particulier, il était de faire progresser les systèmes de suspension pour autocars, c'est-à-dire:

- chiffrer les avantages que peuvent apporter des systèmes avancés de suspension par rapport aux systèmes d'utilisation courante en Amérique du Nord;
- affiner le modèle représentant l'interaction suspension-véhicule et en faire un instrument de conception;
- valider ce modèle par des essais routiers sur des autocars modernes classiques; et
- établir des prévisions chiffrées sur l'apport des systèmes avancés de suspension, y compris la suspension active, à la rentabilité des autocars de type avancé ou courant.

L'approche utilisée pour la recherche a été celle de la modélisation d'un système de suspension, lequel a débouché sur un instrument de travail permettant de prendre en compte soit un nombre plus élevé de composants, soit une configuration complètement remaniée, et d'en tirer des prévisions fiables sur les avantages comparés de chaque composant.

La première étape a donc été une recherche documentaire sur l'état d'avancement de la technique des suspensions d'autocar en Amérique du Nord et en Europe: méthodes d'étude et d'analyse ainsi que revue des types de suspension équipant divers poids lourds, à la recherche d'idées neuves adaptables aux suspensions pour autocars.

Cette étape a débouché sur un petit nombre d'idées intéressantes, à analyser par des moyens informatiques. Ce sont:

- suspension avant à roues indépendantes;
- suspension arrière modifiée à cadre en H pour une plus grande raideur en roulis;
- suspension arrière modifiée à cadre en A pour une plus grande raideur en roulis; et
- suspension adaptative; par exemple, amortisseurs de chocs à réponse adaptative.

Une recherche sur la suspension active a montré que, à terme plus ou moins rapproché, une suspension de type totalement actif ne conviendrait pas à un autocar, pour des raisons surtout fondées sur la fiabilité d'exploitation, la maintenabilité et le coût de première acquisition.

Les caractéristiques de fonctionnement et les performances à exiger des suspensions de type avancé envisagées pour les autocars ont été définies en s'inspirant de diverses sources, et des objectifs généraux de type lignes directrices ont été fixés pour l'étude de suspensions avancées, susceptibles de rehausser la rentabilité des entreprises d'autocars.

Ensuite, ont été élaborées les spécifications techniques pour un programme informatique susceptible de devenir un instrument de conception aussi polyvalent que possible et capable de dégager les critères de performances nécessaires. Avec ces spécifications, et tirant parti de méthodes d'analyse courantes, les techniques de prévision par modélisation les plus appropriées ont été choisies et adaptées à l'étude d'un système avancé de suspension pour un autocar de quelque modèle que ce soit.

Cette deuxième étape a débouché sur la sélection d'un petit nombre de modèles de simulation répondant aux besoins définis. L'option d'élaborer un logiciel spécifique aux autocars et totalement nouveau a été écartée en raison de l'existence de logiciels appropriés, d'autant plus qu'il aurait fallu établir les équations fondamentales par des moyens non automatisés. Les autres options en présence étaient les suivantes:

- Jeu de programmes informatiques élaborés par l'UMTRI pour la modélisation dynamique d'ensembles routiers de diverses configurations. Dûment validés et tombés dans le domaine public, ces programmes ont maintenant plusieurs utilisateurs, notamment le CNR et l'Université Concordia qui se servent de variantes.

La suspension étudiée est subdivisée en groupes de composants dont les caractéristiques sont paramétrées comme s'il s'agissait d'un seul et même composant. Ces programmes servent à des prévisions concernant la directibilité seulement, et nullement la tenue de route. Les données d'entrée concernent des véhicules routiers existants et proviennent d'essais statiques en vraie grandeur visant à déterminer le comportement global de suspensions montées en place. Ces programmes ne conviennent d'aucune façon à la modélisation de systèmes complexes dont on veut étudier le comportement par le détail.

o Trois programmes de calcul automatisé permettant d'établir et de résoudre les équations représentant des systèmes dynamiques non linéaires, tridimensionnels et multicorps, et dont il aurait fallu en choisir un. Le premier de ces programmes était le progiciel ADAMS élaboré par la MDI et qui trouve des utilisateurs de plus en plus nombreux aux Etats-Unis, à cause de son aptitude à réaliser des simulations d'une complexité et d'un niveau de détail inaccessible par des moyens de calcul non automatisés. Mais il pèche par un coût très élevé, environ 40 000 $ U.S. et une très haute technicité. Les prévisions que ce progiciel a permis d'obtenir sur la directibilité des véhicules routiers sont jugées généralement bonnes.

Quant aux deux autres programmes considérés, il s'agit du MEDYNA élaboré par DFVLR en Allemagne fédérale et du RESTRI, un produit de l'Université Queen's. Le premier, dont le prix égale celui du progiciel ADAMS, a surtout servi dans le domaine ferroviaire, là où la modélisation des systèmes de guidage flexibles est capitale; ce programme est le seul à pouvoir représenter le comportement de corps élastiques. Son défaut est qu'il nécessite un matériel de calcul puissant et un personnel hautement qualifié. Le second, le RESTRI, se compare au premier, sauf qu'il n'est pas encore tout à fait au point. Il donne de bonnes approximations concernant des systèmes dynamiques simples, ferroviaires ou autre. Bien qu'il nécessite des moyens de calcul plus modestes, il faut de la part des utilisateurs une initiation longue et ardue.

o Le progiciel A'GEM produit à l'Université Queen's c'est un programme de calcul automatisé capable de résoudre les équations du mouvement dans le cas de véhicules ayant des caractéristiques géométriques linéaires (variations angulaires très faibles), mais qui comprend des éléments présentant un comportement non linéaire. En plus d'être hautement interactif, il permet de représenter de gros systèmes sur matériel informatique de petite puissance. Cependant, s'il permet d'établir les équations de base, il ne peut pas les résoudre, cette fonction devant être assurée par un autre programme, conçu exprès; or, UTDC possède un logiciel capable, avec peu de modification, de prendre en charge ces calculs. Le coût d'acquisition du A'GEM est de 1 200 $ pour le code source et de beaucoup moins sous forme de progiciel.

A partir de ces renseignements, et connaissant les caractéristiques que doit posséder l'instrument de conception, il a été décidé d'opter pour le programme A'GEM, complété du logiciel spécifique dont dispose UTDC. Les programmes UMTRI ont été écartés parce qu'ils n'allaient pas assez loin dans le détail et qu'ils ne permettaient pas d'obtenir des prévisions sur la tenue de route. Bien qu'il eût été intéressant de choisir un programme permettant de représenter des systèmes non linéaires, comme le programme ADAMS aurait pu le faire vu le degré de réalisme qu'il autorise, des facteurs tels le coût d'acquisition et d'entretien, l'achat du matériel nécessaire et la compétence dont doit faire preuve le personnel d'exploitation ont joué contre cette option.

Le progiciel qui a finalement été retenu se présente sous la forme d'un ensemble de modules, faciles d'utilisation par le projeteur et capables de fournir en sortie l'ensemble des paramètres variables recherchés. Etant donné que le modèle ainsi réalisé comportait des éléments de suspension à comportement non linéaire, la modélisation restait dans le domaine temporel. Or, la directibilité étant représentée dans le domaine temporel, aucun post-traitement des résultats n'a été nécessaire. Par contre, les prévisions sur la tenue de route sont généralement exprimées dans le domaine fréquentiel; il a donc fallu procéder à un post-traitement, celui des valeurs d'accélération qui varient dans le temps, tout comme on l'aurait fait s'il s'était agi de données tirées d'expérimentations pratiques. Le logiciel capable de résoudre les équations et de faire le post-traitement, propriété de l'UTDC, a été modifié pour l'intégrer au progiciel de conception envisagé.

Des essais routiers ont été faits avec un autocar classique circulant sur un parcous établi d'avance et dûment instrumenté; ces essais avaient pour but de caractériser des suspensions existantes, en faisant varier leurs caractéristiques de réponse en configurations normales et altérées. Les variations de comportement d'une configuration à l'autre ont permis de valider le mode de simulation, par comparaison des résultats réels aux résultats théoriques fournis par le modèle.

L'autocar utilisé était un modèle 102-A3 fourni par le fabricant, la compagnie MCI; il a été soumis à une batterie d'essais en régime permanent et transitoire afin d'en caractériser la directibilité et la tenue de route dans des conditions contrôlées. Les autres paramètres pris en compte ont été le poids, le centre de gravité et l'influence de l'essieu médian.

Ces essais ont permis d'affiner le modèle et de donner aux prévisions qui en sont issues un niveau de confiance élevé. Par la suite, ce modèle ainsi que le programme d'analyse ont servi à caractériser, sous l'angle de la faisabilité et des performances, des systèmes de suspension améliorés, des modifications éventuelles à apporter à des systèmes existants et des systèmes auxquels des éléments de suspension active ont été adjoints. Les avantages de tous les systèmes envisagés ont été déterminés et le plus prometteur d'entre eux a fait l'objet d'une recommandation de mise au point en vue d'en équiper les autocars nord-américains.

Les conclusions auxquelles a abouti la recherche peuvent être résumées comme suit:

- Bien que les autocars de fabrication nord-américaine soient équipés de suspensions robustes, durables et adaptées aux conditions routières, des améliorations dans la directibilité et la tenue de route sont souhaitables et réalisables.
- L'exemple de l'Europe montre que d'importants progrès peuvent être réalisés en apportant des améliorations à la fois simples et classiques.
- On ne prévoit pas qu'une suspension totalement active puisse devenir économiquement intéressante dans un avenir plus ou moins approché. Dans l'état actuel de la technique, ces fortes réserves sont motivées par des considérations de fiabilité, maintenabilité et sécurité.
- Les résultats obtenus par modélisation concordent assez bien avec les résultats pratiques des essais routiers de l'autocar MCI 102-A3, tant en configurations normales qu'en configurations altérées, mais les variations paramétriques testées ont eu des effets plus faibles que prévu.
- Tout porte à croire qu'il sera nécessaire de modéliser la raideur de la carrosserie pour obtenir une représentation plus fidèle du comportement réel du véhicule.
- Les composants du châssis ayant eu la plus forte influence sur les résultats ont été les amortisseurs de chocs. Ceux de l'autocar testé, il est important de le dire, ont affiché un comportement très inégal.
- Pour une représentation plus fidèle, la non-linéarité des amortisseurs de chocs ainsi que l'amortissement de frottement doivent être modélisés.
- La régulation active de l'amortissement est un procédé réalisable à bon compte qui peut améliorer la maniabilité sans altérer la tenue de route en marche normale.
- La modélisation a montré que, dans le cas d'un autocar de type avancé, les meilleurs résultats seraient obtenus avec une suspension avant à roues indépendantes, un essieu porteur médian et un pont moteur arrière à cadre triangulaire.
- Il semble impossible d'améliorer sensiblement la suspension sans réétudier l'architecture de l'autocar au complet.
- D'autres améliorations appréciables peuvent être apportées par un système de suspension à cadre triangulaire ou en H dont l'assiette aura été élargie, de manière à augmenter la raideur et l'amortissement en roulis.

Les recommandations suivantes sont faites:

- Le programme de modélisation devrait comprendre des équations représentant ne serait-ce qu'une partie de la raideur de la carrosserie (le programme actuellement au point en est parfaitement capable).
- A court terme, il faudrait penser à augmenter la raideur et l'amortissement en roulis de la suspension arrière.
- A long terme, il faudrait axer la recherche sur un système comportant une suspension avant à roues indépendantes, un essieu porteur médian et un pont moteur arrière à cadre triangulaire.

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1.0 INTRODUCTION

1.1 BACKGROUND

The work described in this report consisted of an indepth analytical study of advanced suspension systems for highway buses. This study ensued in part from work undertaken by UTDC Inc. (UTDCI) for the Transportation Development Centre (TDC) to complete the conceptual design work required to manufacture a Canadian, advanced technology double-deck intercity coach (1).

An improved suspension system is crucial to advancing the state-of-the-art in bus design. This requirement was identified by UTDCI, by firms in the bus industry, and by the ad-hoc committee organized by TDC to provide advice and direction on projects in the intercity motor coach industry.

The project takes advantage of previous work in this field by UTDCI and other contractors to TDC. The work is complementary to work done by the National Research Council (NRC) in truck suspension/ parametric measurements and by Concordia University in the seat ride vibration analysis.

1.2 OBJECTIVES

The overall objective was "to contribute to more efficient and attractive highway bus transportation and thereby improve the health of the carrier industry, reduce total transportation costs and encourage greater public use of this economical and energy efficient mode" (2).

The sub-objective was to improve conventional highway buses by incorporating new technological developments and systems innovations in current and future designs.

This project addresses a crucial subcomponent area to make advances in the state-of-the-art with respect to bus suspension system design. The specific objectives of this project were to:

- quantify the benefits of advanced suspension systems compared to existing North American suspension systems on highway buses.
- improve and generalize the suspension/vehicle interface model as a design tool;
- verify the model by means of road vehicle testing using a typical modern conventional highway bus; and
- make quantitative predictions of the benefits of applying advanced suspension system concepts, including active suspension concepts, to both existing and new design highway buses.

1.3 OVERALL METHODOLOGY

The project approach to the overall objective of improving highway bus transportation was to address the design modelling aspect of bus suspension. A design tool was developed in which additional components and restructured suspension configurations can be easily examined to predict, with a high level of confidence, component-level design improvements.

The first step of this process was to study state-of-the-art technology in highway bus suspension, particularly in North America and in Europe. Included in the study were current methods for design and analysis, and a review of other heavy vehicle suspensions for new technical ideas and adaptation to highway buses.

Operational needs and performance requirements, based on a variety of sources, were established for the suspension system of a highway bus, and general targets set as guidelines for advanced designs that would improve or retain economic viability for the operator.

Specifications for a general computer model and analysis of highway bus suspension performance were then formulated, ensuring that the necessary performance criteria be predicted and that sufficient flexibility be incorporated for use as a design tool. Based on these specifications and on the methods of analysis currently available, the most appropriate modelling techniques were selected, modified and adapted to the development of an advanced computer suspension model for any highway bus.

Real suspension performance data was acquired by instrumenting and operating a conventional highway bus over a predetermined course and varying the response characteristics of the bus suspension system in both on and off-design configurations. The measured variations in performance allowed verification of the suspension model through comparison of computer predictions with actual data.

The suspension model was further developed and refined until a high level of confidence (credibility) was achieved in the computer predictions. The model and computer analysis were extended to examine the feasibility and performance expectations of improved suspension system designs, potential modifications to existing designs and of the incorporation of active suspension elements into the designs.

Performance benefits were identified for each suspension design, and recommendations made for the development of the most promising suspension system for highway buses in North America.

The relationship between these tasks is illustrated in Figure 1, Task Network.

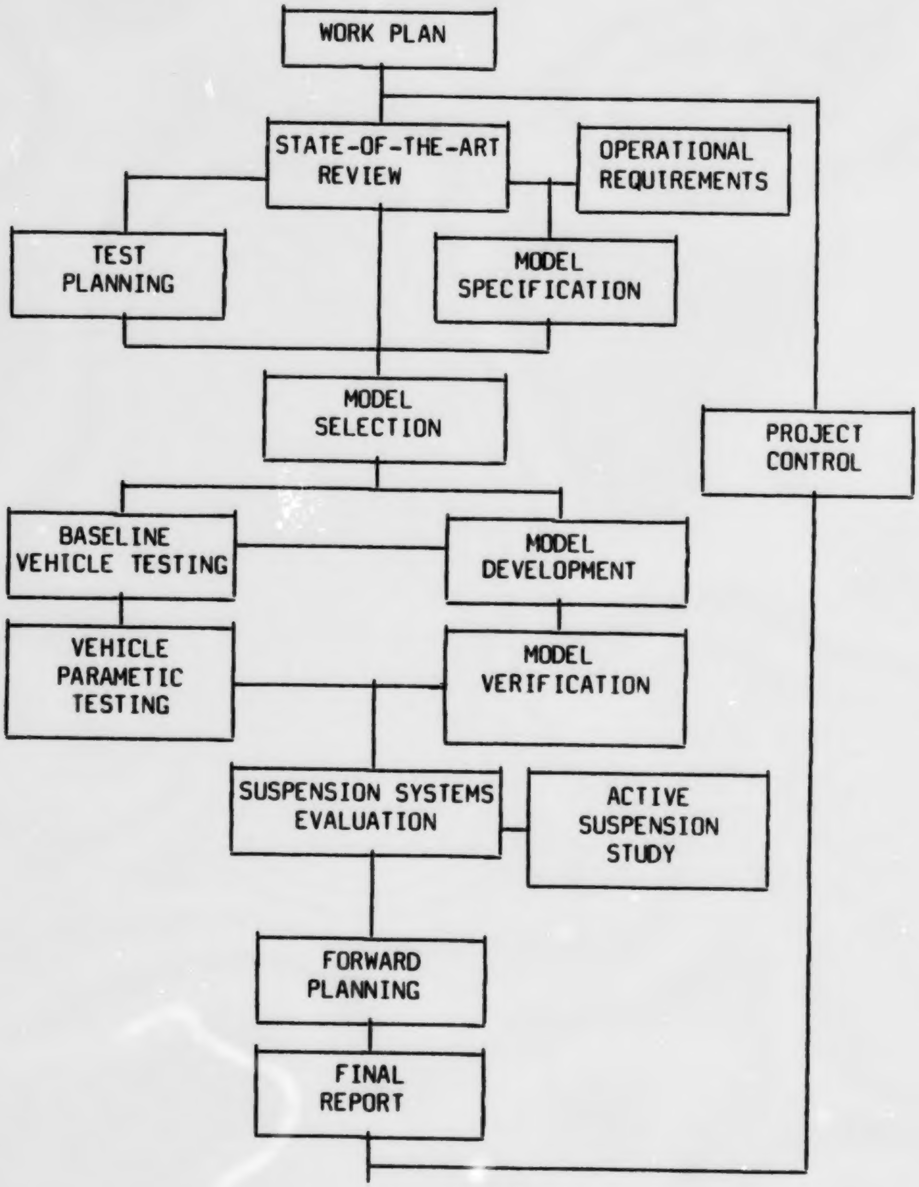

WORK PLAN
STATE-OF-THE-ART REVIEW
OPERATIONAL REQUIREMENTS
TEST PLANNING
MODEL SPECIFICATION
MODEL SELECTION
PROJECT CONTROL
BASELINE VEHICLE TESTING
MODEL DEVELOPMENT
VEHICLE PARAMETIC TESTING
MODEL VERIFICATION
SUSPENSION SYSTEMS EVALUATION
ACTIVE SUSPENSION STUDY
FORWARD PLANNING
FINAL REPORT

FIGURE 1: TASK NETWORK

2.0 STATE-OF-THE-ART REVIEW

The state of the art review carried out was documented extensively as an interim report (3). In this section the review is summarized.

2.1 SUSPENSION SYSTEM PERFORMANCE REQUIREMENTS

The suspension system comprises all elements required to suspend the coach body from the wheel assemblies (unsprung mass). In carrying the vehicle load, the suspension must impart acceptable ride quality to the vehicle body while retaining good stability and handling characteristics. Suspension elements include springs, dampers (shock absorbers), torsion bars, stabilizers, control arms, links, ball joints, etc. The suspension positions the coach body with respect to the axles. Related subsystems, whose design must be specifically included with the suspension system components are the wheels and tires and the steering assemblies.

In terms of performance, the ideal suspension system isolates the Coach body from road irregularities to give good ride quality, yet responsively transmits the vehicle control forces of steering, braking and traction between the body and the ground without excessive body disturbance to give good handling. The suspension should enhance the ability of the pneumatic tires (which ultimately support and guide the vehicle) to maintain contact with uneven road surfaces and maintain correct orientation relative to the vehicle and to the road surface.

The following are requirements common to the suspension of a conventional intercity coach.

o Suspension elements should be chosen and distributed in such a manner as to provide, if possible, equal vertical loading on each wheel in any loaded condition.

o Axle and tire loadings must not exceed the legal regulations and standards. Hence, the number of axles and tires for a vehicle being designed can be initially established; suspension elements and pick-up points must then be such that the limits are not exceeded in any loaded condition.

o Suspension elements must be capable of absorbing operational loads due to steering, acceleration and braking and also provide for contact with curbs and minor off-road excursions without damage.

o The vertical travel of the suspension system elements (and hence the vertical travel of the coach chassis relative to the axles) under all static and dynamic operating conditions must be limited to prevent interference of the chassis with other coach subsystems and with the roadway.

o The suspension should provide vehicle ride comfort, both in vertical and horizontal directions, meeting the International Standards Organization (ISO) "reduced comfort boundary" for a trip duration of eight hours; viz. all random or discrete frequency vibrations experienced by the passengers as a result of Coach over-the-road operation and on-board equipment, both horizontally and vertically, should meet the ISO Standard (4) for an eight hour ride (the 95th percentile trip duration for intercity service is taken as eight hours).

- To produce acceptable ride, suspension natural frequencies must be kept low, in the order of 1 Hz.
- The suspension must provide vehicle stability in all directions and adequately control lateral, longitudinal and torsional movements at each axle.
- Vehicle handling performance is a complex area which is difficult to quantify but reflects directly on passenger comfort and safety. Optimal handling characteristics include good steering and braking control without excessive body motion (best achieved with a firm suspension system).
- The suspension elements must be compatible and not interfere with the coach steering. In particular, the coach must turn within a dynamic envelope that is not hazardous or disruptive to other road users; a maximum turning radius of 12.5 m (41') is considered acceptable. Hence, front suspension elements must permit adequate steering lock angles to meet this requirement.
- Good reliability of the suspension elements is essential; failure of components is potentially dangerous and particularly visible; if possible, suspension elements should be designed for the lifetime of the coach (15 to 20 years).
- Scheduled maintenance of suspension system elements should be minimal, and certainly should not be required at intervals of less than one month or 18,000 km.
- The ability of the suspension system to meet the performance requirements must be balanced by the overall cost of necessary components; obviously minimizing cost is important.
- The body height in relation to the centre line of the wheels must remain constant to within 12.7 mm (0.5") in the static condition, regardless of load, if underbody clearances are to be maintained with allowance for dynamic travel.
- The vertical travel of the suspension elements should be limited to a maximum of 0.102 m (4") in either compression or rebound.
- Typical suspension vertical design capacities range from 4-5 g to allow for dynamic loads, and thus all suspension components must be capable of withstanding these design loads.

- In the horizontal directions, the suspension must be capable of accepting loads of 0.6 g under normal operation and up to 2 g for shock loads.

2.2 SUSPENSION DESIGN, NORTH AMERICAN COACHES

2.2.1 MCI/TMC

MCI/TMC is currently the largest intercity bus manufacturer in North America. The four MCI models currently in production have very similar suspension systems. The drive axle suspension on all models is identical; however, the front suspension can have either 4 or 2 air springs depending on the model.

The front suspension of the earlier 3 axle buses (MC-9, 96A3) are all air beam type consisting of four 9" convoluted bellow air springs mounted on an H-frame with the springs not too widely spaced. Two shock absorbers are fitted on the front axle. Position of the front axle is through the use of four parallel trailing links and one transverse radius rod (panhard rod) (See Figure 2).

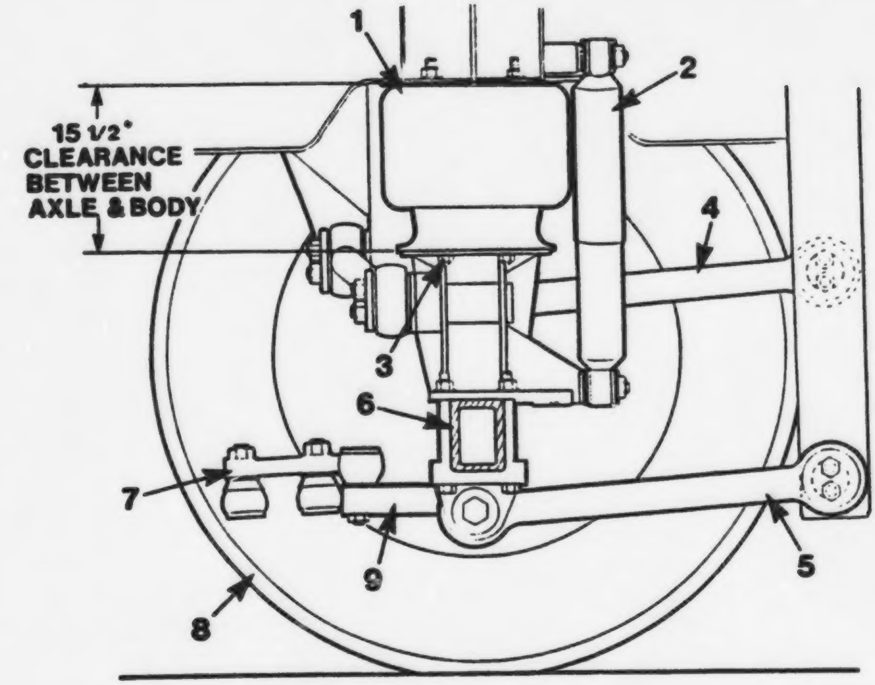
The dual tire drive axle suspension of all MCI models is identical. The suspension is constructed of an H-frame beam axle with four 12" bellows, also not very widely spaced, however, a roll stabilizer bar is added to the rear suspension to increase roll stiffness of the bus. Four inboard shock absorbers are also added to the suspension. Positioning of the drive axle is by 2 semi-leading links and 2 leading links.

The third axle of all three axle MCI buses is a tag axle. The wheels are independently suspended on relatively widely spaced trunnion mounts and are sprung by one 11" reversible sleeve air spring per wheel. Also, one shock is mounted per wheel. The tag axle contributes significantly to the buses' roll stiffness.

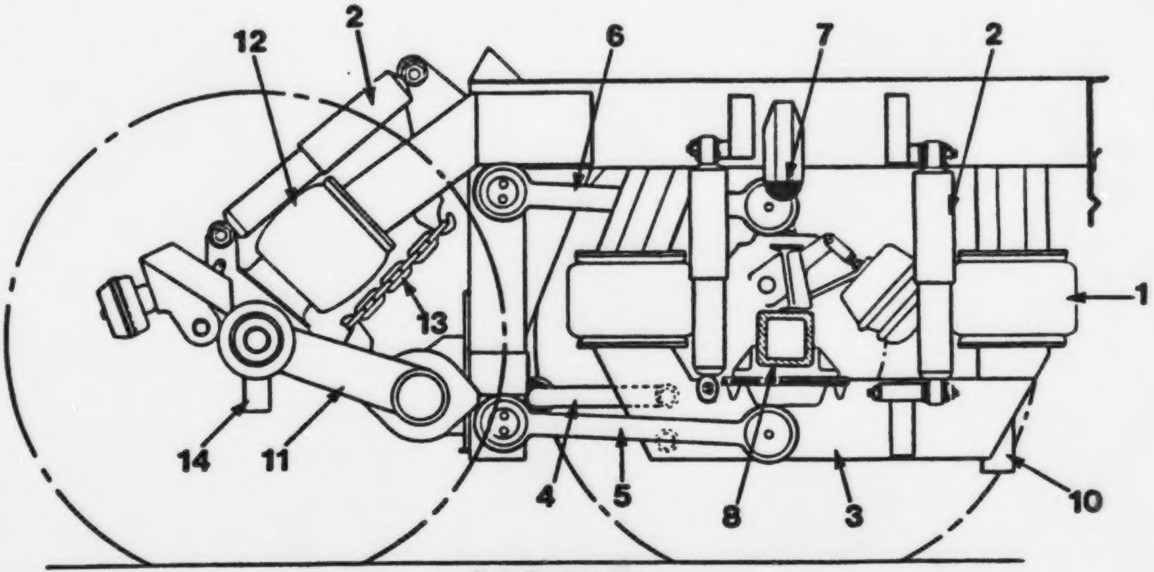
For more recent 102A3's, 96A3's and all 2 axle buses, the front suspension has evolved to an air beam or beam axle with only 2 air springs and 2 shock absorbers.

In general, MCI buses have reasonable ride quality; however, there may be room for improvement, specifically in the areas of roll stiffness and roll damping.

1. Air Spring
2. Shock Absorber
3. Nut
4. Upper Radius Rod
5. Lower Radius Rod
6. Axle
7. Axle Arm
8. Tire
9. Pitman Arm


1
2
15 1/2"
CLEARANCE
BETWEEN
AXLE & BODY
4
3
6
7
9
5
8

Front Suspension


12
2
6
7
2
1
13
14
11
4
5
8
3
10

Rear Suspension

1. Air Spring (Drive Wheel)
2. Shock Absorber
3. Suspension Support
4. Anti-Sway Bar
5. Lower Radius Rod
6. Upper Radius Rod
7. Axle Bumper
8. Axle
10. Jacking Point (Drive Axle)
11. Trailing Axle Arm
12. Air Spring (Trailing Wheel)
13. Hold-up Chain
14. Jacking Point (Trailing Wheel)

FIGURE 2. MCI 102-A3 SUSPENSION

2.2.2 Prevost Car, Inc.

Prevost is in third place in North American bus sales, closely following Eagle when it comes to production quantities. The suspension of all Prevost bus models (except the H5-60) are identical.

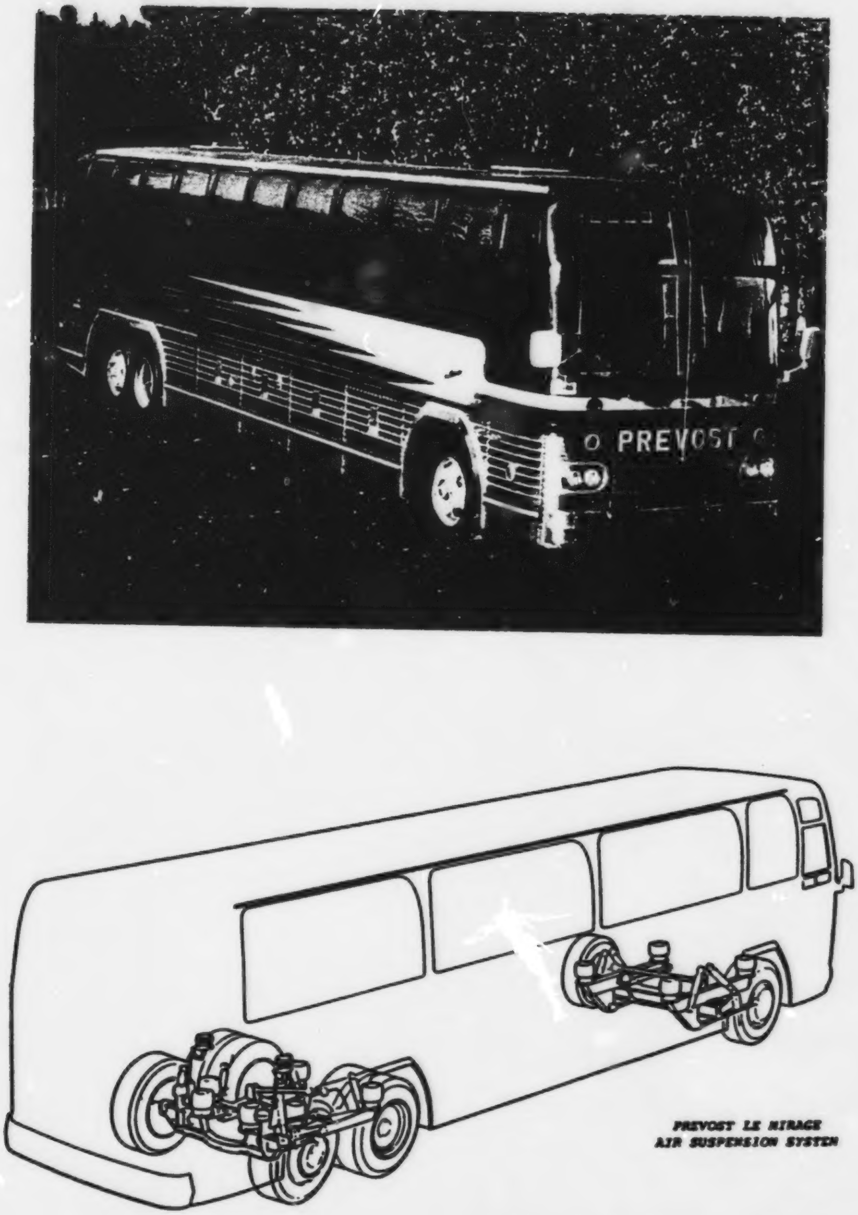
The front suspension consists of an air beam/H-frame with 4 rolling sleeve bellows and 2 shock absorbers. Positioning is through the use of 4 trailing links and one panhard rod (See Figure 3).

The drive axle suspension of the Prevost Bus is also constructed of an H-frame. The arms of the H-frame are especially long so that the 4 rolling sleeve air springs can be mounted on a wide base, thus giving good roll stiffness to the rear suspension. As well, four shock absorbers are used. Position of the axle is by 4 trailing links and a panhard rod.

The third axle of the Prevost bus is a beam "tag" axle. This axle is essentially a beam axle with no H-frame superimposed. The two rolling sleeve air springs are mounted on a relatively narrow base, while the shock absorbers are outboard of the springs. Positioning of this axle is by 4 trailing rods and a panhard rod.

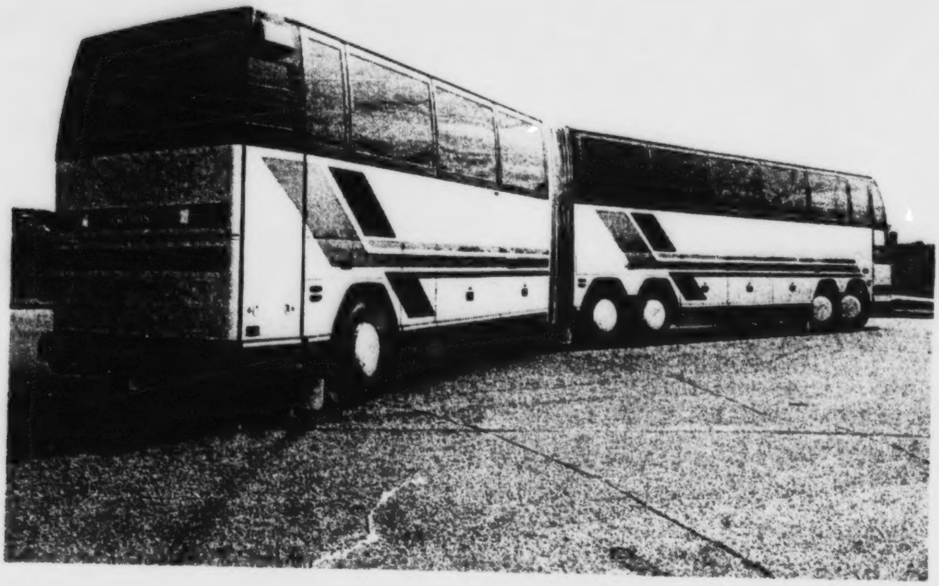
No roll bar is mounted anywhere on the bus, thus raising the question of the vehicle's overall roll stiffness. The springs on the drive axle are mounted on a wide base, which explains the logic for omitting the roll bar. There is no conclusive evidence available to show how this bus compares to other buses in terms of ride and handling.

Prevost's latest intercity bus is the H5-60 (Figure 4); an articulated 5 axle bus with a seating capacity of 80 people. A pair of beam axles is used at the front of the bus with all 4 front wheels steering. The suspension of both front axles is by 2 air springs per axle and 2 anti-roll bars. Positioning of the front axles is by 4 trailing links and a panhard rod on each axle. Two drive axles are used, each consisting of rigidaxles with 2 single wheels mounted on each axle. Again, springing is by 2 air springs per axle, plus a roll stabilizer bar and positioning here is by 2 leading/2 semi-leading links. The back axle (axle on articulated unit) is a beam axle with wheels which steer. Springing is by two air springs and a roll stabilizer bar, while positioning is by 4 trailing links.


PREVOST
PREVOST LE MIRAGE
AIR SUSPENSION SYSTEM

*A "tag" axle is a non-driven axle located behind a driving axle.

FIGURE 3. PREVOST LE MIRAGE



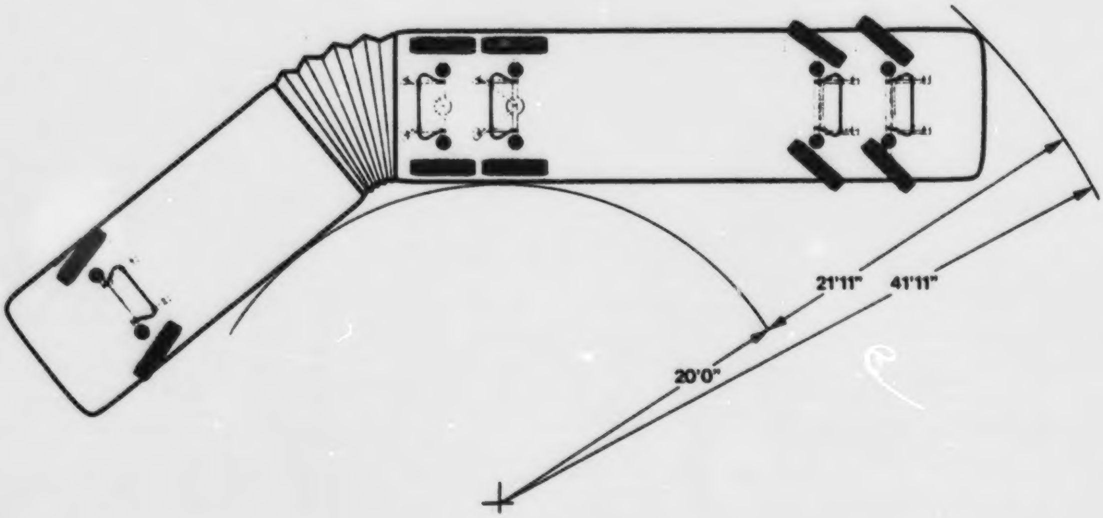

21'11"
41'11"
20'0"

FIGURE 4. PREVOST H5-60

2.2.3 Eagle International, Inc.

Eagle produces essentially two models of buses (Golden Eagle II & American Eagle Model 10) and both buses use the same "Torsilastic" suspension.

The front suspension is an independent Torsilastic suspension with upper and lower control arms. The rubber torsion "tubes" are rigidly connected to the bus frame and act as the pivot axis for the lower control arm as well as providing the springing torsion for the bus front suspension. Damping is provided by two shock absorbers (Figure 5).

The drive axle suspension uses a beam axle on which two Torsilastic units are rigidly mounted parallel with respect to the buses' long axis (Defined by the Society of Automotive Engineers as the X-axis). Torsion arms project off both ends of these tubes and fasten to the bus frame at very widely spaced points, thus providing an excellent roll stiffness contribution from the rear suspension. The torsion tubes also provide the positioning for the drive axle. Four shock absorbers are mounted.

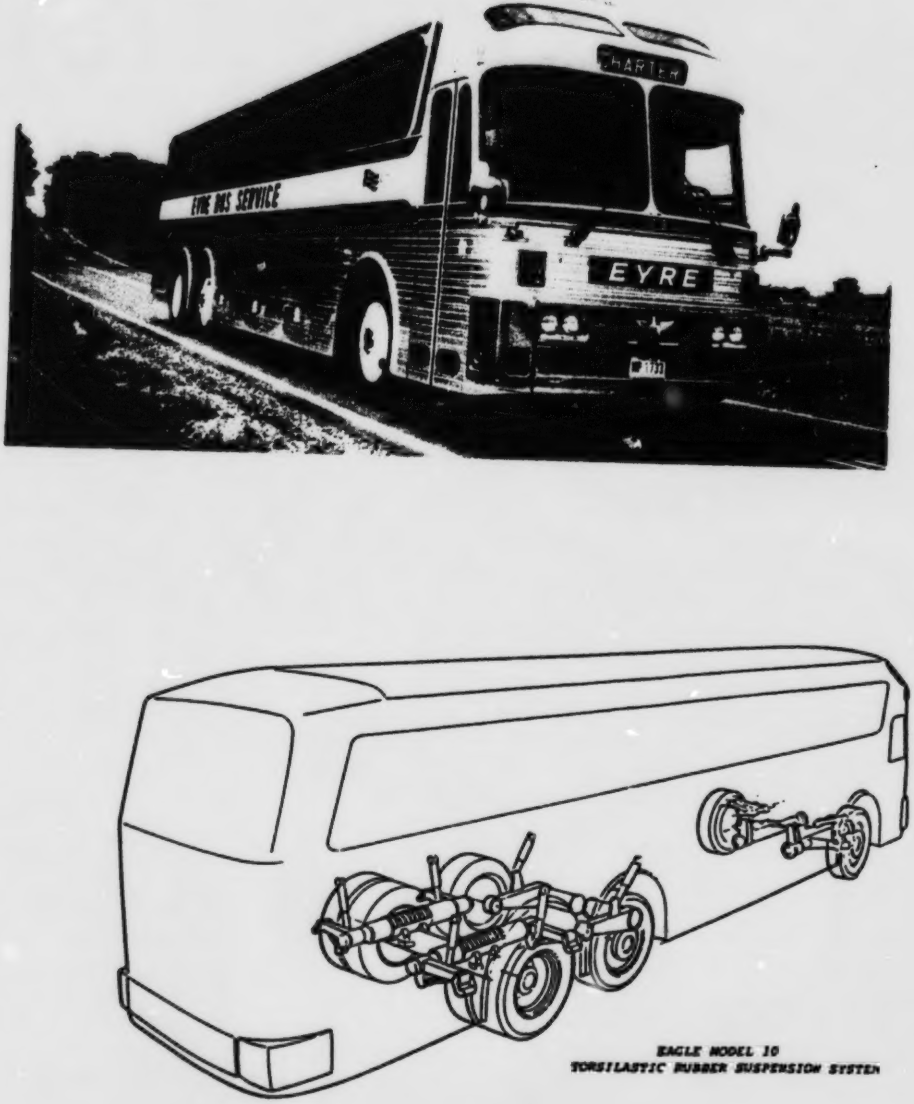
The third axle is an independent "pusher" axle. Each wheel is independently sprung by a Torsilastic tube which is rigidly mounted to the bus chassis. One shock absorber per wheel is used.

The overall ride characteristics are claimed to be quite good for the Eagle. Due to the very wide effective centres of the springs, the roll stiffness of the Eagle is quite good. As well, Eagle claims that the Torsilastic suspension provides a "better ride" than MCI. However, a disadvantage to the torsilastic suspension is in its inability to self-level. This means that the bus rides high when unloaded and much lower when fully loaded.

2.3 SUSPENSION DESIGN, EUROPEAN COACHES

Coach design in Europe has moved rapidly towards the high floor, high capacity end of the market, with many manufacturers offering 66 seat single deck high floor and either 72 or 84 seat double deckers.

Suspension and vehicle design for stability, handling and braking with these vehicles is considered to be a major problem, and the E.E.C. (European Economic Community) is considering setting up a special engineering committee to examine possible legislation in this field. Little has been published in the field of handling measurements on coaches, and there is no clear indication that independent front suspensions are preferred by drivers, purchasers or their customers.


CHARTER
EYRE BUS SERVICE
EYRE

EAGLE MODEL 10
TORSILASTIC RUBBER SUSPENSION SYSTEM

FIGURE 5. EAGLE WITH TORSILASTIC SUSPENSION

2.3.1 General Suspension Characteristics

Major European manufacturers universally use a beam rear axle, but there is a definite split in front axle design. Manufacturers such as Mercedes, Volvo, Daf, Scania and Leyland retain beam axles, while Setra, Neoplan, MAN, Renault use double wishbone independent systems. Van Hool use independent front suspension on their 4 m high vehicles and beam axles on other models. Rolling lobe air suspension with double acting telescopic dampers are used almost universally.

- Rear Suspension

  The most common rear axle location and springing is the 'H' frame, with the four rolling lobe air bags mounted outboard, on or around the inner wheel centre line and the damper mounted outboard of the spring. Fore and aft and lateral location is usually either by four trailing links and a panhard rod or by a 'A' frame upper link and two lower trailing links. Torque reaction is also taken by these links. Volvo use an upper 'A' link and two lower links, which are leading (associated with an engine/transmission in front of the rear axle). Leyland Tiger and Duple 425 use a single leaf WEWELER mounted spring to locate the 'H' frame. Neoplan, Daf, Leyland and Volvo on their B10R use different forms of trailing 'A' frame. Neoplan and Volvo B10R use a true 'A' frame with the air springs and dampers mounted behind the rear wheels at a leverage of between 0.7 and 0.8 to 1. Daf and Leyland Lynx use two trailing single leaf springs clamped directly to the axle which are then continued behind the axle to pick up a cross beam on which the air springs and dampers are mounted. For roll the leaf springs are forced to twist giving additional roll stiffness. Daf use a panhard rod which is bent requiring the use of a solid 50 mm square bar. Leyland have no additional lateral location relying solely on the lateral stiffness of the forward spring mounting bushes and the stiffness of the clamped joint between the axle and springs.

- Front Suspension

  - Independent Front Axle

    All the major independent suspension designs are of the double wishbone type with 'A' frame upper and lower arms. Rolling lobe air suspension is usually mounted directly onto the vertical link, thus not significantly loading the arm bushes from the vertical wheel load. In addition the damper is also mounted to the lower part of the vertical link, again not significantly loading the arm bushes from damper loads.

Most independent systems use a swivel pin which defines the King Pin Inclination, offset and steering geometry, with separate rubber bushed arm pivots for vertical freedom and compliance. The advantage of this system is that the very expensive, well developed casting/forging of the hub is retained from previous beam axle systems in production. The exception is Renault who use ball joints top and bottom for both steering and vertical movements and load the top ball with the air spring and the bottom ball by both the damper and an additional small coil spring.

There is large variation in arm lengths, angles and therefore, effective swing axle lengths and roll centre heights.

Castings, forgings and tubular fabrications are used. Rubber bushes are used universally for arm pivots, except in the G.K.N. proposal which uses taper roller bearings.

- Beam Front Axle

Location is mainly by trailing links, and a panhard rod or 'A' frame for lateral location with a single rolling lobe air bag together with single or twin telescopic dampers mounted inside each wheel on the axle centre line. This layout is used by Mercedes, Volvo B10M, C10M and Daf.

Leyland, on some models, use an 'H' frame located by leaf springs allowing the four air bags and dampers to be mounted outboard and on the Lynx use a similar location to that described at the rear, with two trailing links formed from single leaf springs and the air bags mounted outboard behind the wheels. A panhard rod is used at the front for axle lateral location.

The Volvo B10R uses an 'A' frame with panhard rod.

o Tag Suspension

Most manufacturers use their front beam or independent suspension system with the steering locked. Some (Neoplan) have a special hub fabricated with no steering freedom.

Setra use a "dead" beam axle of a large diameter tube located by links.

2.3.2 Design Trends

There has been a fairly rapid drive towards independent suspension at the front by some manufacturers in Europe, accelerated by the move to double deck versions which require the lowest possible floor and high roll stiffness (Figure 6). Nevertheless, Daf, Scania, M.C.W. and Leyland all produce successful double deck designs with beam front axles. The two high volume, high quality manufacturers of Mercedes and Volvo have also retained beam axles with very satisfactory performance on high floor single and semi-double deck vehicles.

Significant reference should be made to the new Volvo C10M which was announced in 1984 and is just starting volume production (approximate maximum of 1000/year which is high by European standards). This Coach uses a beam front axle and has just won the prize for the best technical performance (acceleration, hill climb, stability, braking, noise level and manoeuverability) at the Belgian KORTRIJK show, voted by technical journalists, and also Coach of the Year by the same judges.

It is believed unlikely that Volvo and Mercedes, two very technically competent companies, will change to independent front suspension within the next five years on intercity coach designs.

2.3.3 Independent Front Suspension Compared With Beam Axle

- Unsprung Weight

  The main advantage claimed for independent suspension systems is a reduction of unsprung weight. However, on a coach 6½ tonne axle layout, the unsprung weight is dominated by the same wheel, tyre, brake, hub bearing and swivel assembly which is used on both beam and independent systems.

- Roll Stiffness

  The total roll angle of European coaches with beam and independent suspensions are similar. This is achieved by the addition of massive (50 mm diameter) anti-roll bars for the beam axle compared with no anti-roll bars, or small additional roll-bars, with independent systems.

- Roll Damping

  There is a significant gain in roll damping with independent systems compared with beam axle solutions. The damping which is set for vertical control acts at the effective track on independent systems, whereas on a beam axle they normally can only be mounted about ½ track apart. As this roll damping leverage appears as a 'squared' effect, the roll damping on the beam versions is only approximately 25% of a similarly damped

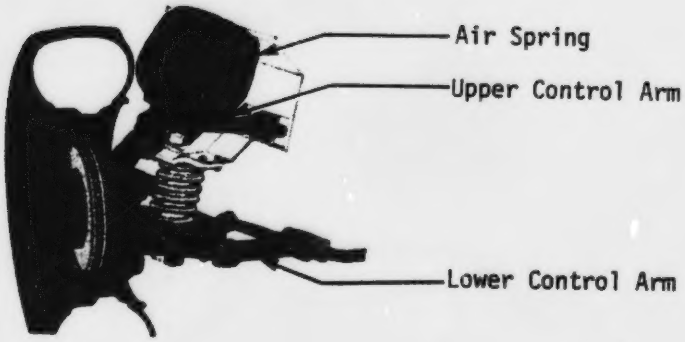

Air Spring
Upper Control Arm
Lower Control Arm

FIGURE 6: INDEPENDENT FRONT SUSPENSION

independent suspension. This has both lateral acceleration (ride) and stability implications and should be investigated further on both the ride and handling computer simulations.

- Geometry Control

  Most European independent suspension systems have retained the same swivel/hub arrangement that was developed and used for beam axles. Thus the King Pin Inclination (K.P.I.) and offset have been historically defined, with low K.P.I. (5-7 degrees) and high offset (60-120 mm). This high offset can cause problems with steering feedback from the road, high steering torques from a tire blowout, or a single brake failure/brake unbalance.

  Maximum lateral tire forces of the wide (295-315 mm) commercial vehicle tires used on coaches are produced at low camber angles to the ground. This is naturally achieved with a beam axle but is more difficult to design into an independent system. This does not seem to have been a major design factor in several European solutions. The steering system must also be carefully designed to minimize bump steer on independent systems. Again some European designs are not good in this respect.

- There are obvious packaging advantages for independent suspension systems, particularly for low floor vehicles such as double deck designs.

Summary

UNSPRUNG WEIGHT:	Slight advantage to independent
ROLL STIFFNESS:	Equal with roll bars
ROLL DAMPING:	Massive advantage to independent
STEERING GEOMETRY:	Potential advantage to independent not generally used in Europe
HANDLING:	Advantage to independent if properly Designed (Changes in handling due to change in vehicle layout can be significant)

2.4 APPLICABLE HEAVY VEHICLE SUSPENSION TECHNOLOGY

The following suspension systems in heavy vehicles are the most relevant to bus suspensions.(5)

- Air suspension is an accepted and well proven concept in the heavy vehicle area. There are however, disadvantages not only due to the air hardware (piping, compressor etc.), but also, in the added complication over leaf springs due to the mechanical design of the suspension. Leaf springs have inherent lateral stability whereas air springs have none. This leads to the requirement, in general, of more/or stronger links for restraining the suspension laterally.

Despite the above disadvantages, air suspensions fulfil many of the suspension requirements for buses. They are relatively compact; are available with low natural frequencies (1 Hz with the correct suspension configuration); and large load variations can be tolerated while maintaining the vehicle at the correct height.

- Significant work has been carried out on hydro-pneumatic suspensions for heavy vehicles. Some of this work, has been carried out on off-road vehicles such as the WABCO off-highway HAULPAK trucks and is not as applicable to bus suspensions, however, other work into active roll control and vibration isolation (6) is very applicable to bus suspensions since the overall goal is focused on ride quality (which is very important to buses).

2.5 APPLICABLE AUTOMOBILE SUSPENSIONS TECHNOLOGY

In all "On-road" vehicle applications there are three primary concerns which stimulate suspension improvements. One is axle loads, another is ride quality and finally there is handling. Unlike bus and truck suspensions, which are concerned with axle loads, handling and ride quality, automobile suspensions are concerned only with ride quality and handling. Thus, it seems logical that the automobile suspension has been given much attention, especially when it comes to applying advanced technologies such as active/semi-active suspensions which are currently ride and handling oriented. As well, the immense production volume of automobiles as compared to buses and trucks permits this type of experimental activity to be economically possible. The following section briefly discusses some of the recent suspension systems which have been implemented on automobiles.

2.5.1 Auxiliary Pneumatic Strut

In recent years, there has been much attention given to an "Electronically controlled variable rate auxiliary pneumatic spring and adjustable shock suspension" (7,8). This is a packaged system which consists of a "MacPherson Strut" type of suspension unit with coil spring, auxiliary air spring and adjustable shock absorber all concentric on a strut; a number of "sensors" which detect various driving demands; and an electronic control unit. Although the components of the suspension are not unique, the way in which the system operates as a whole is. Basically the system has two operating points - hard and soft. In soft mode, the vehicle rides very smoothly. In hard mode, however, the springs stiffen and the shock absorbers become "firmer" thus producing a "firm" ride suitable for rapid cornering and braking (essentially very good handling is achieved). Through detection of such parameters as vehicle speed, steering wheel angular velocity, acceleration (in three directions) and suspension stroke the ECU ("Electronic Control Unit") decides which of the two suspension settings should be chosen. Switching between the two settings (when in "auto" mode) by the ECU is very fast.

One of these systems was produced by Mitsubishi and it is used on cars such as the Ford Eltec and Mitsubishi's Galant Royal sedan. As well, Bosch has independently developed a very similar prototype system.

This system is not described as "Active" but is instead referred to as "Adaptive". The overall ride quality and handling is apparently very good.

2.5.2 Hydro-Pneumatic Strut

A very interesting suspension component used on some automobiles is the hydro pneumatic strut. Although it has been applied to trucks and buses, the primary applications have so far been concerned with automobiles. The basic operation of the hydro-pneumatic strut involves a fluid for height adjustment of the strut and an air/gas chamber for the springing. Damping can also be incorporated into the strut. Generally, operating pressures are high as compared to air springs and thus hydro-pneumatic struts are a step closer to "optimal active suspensions".

Hydro-pneumatic struts have been used on the Citroen; on the Lucas Girling "Active" Suspension System; and by Moulton Industries (9) on several British Leyland Automobiles (as well as on the Moulton Prototype Bus). The Citroen used the strut primarily as a levelling suspension system. Lucas Girling, however, used hydro-pneumatic struts on their "Girling Modified Ride Suspension". The Girling Suspension strut uses nitrogen gas springs and has independent levelling at each wheel. Each strut is wound with an induction coil for position sensing. The system is controlled by a microprocessor which adjusts the suspension according to three modes of vehicle operation:

1) Start-up-suspension height is adjusted very quickly (several seconds);

2) Slow mode (beginning of journey) - height adjustment reaction time is lengthened; and

3) Trim mode (cruising) - height corrections are small and due to long term load changes such as areodynamic effects or weight reduction due to fuel consumption (typically 20 seconds before decisions are made by the microprocessor in this mode).

Another system (a prototype) which uses the hydro-pneumatic strut was designed by Automotive Products Ltd. (UK). This system is described as "active" since it is capable of compensating for vehicle roll and pitch during extreme cornering and braking manoeuvers. The system uses an acceleration sensing mechanism (a mass & spring) to control a hydraulic control valve which regulates fluid continuously to the struts as well as providing levelling. The system was simulated on a computer; in addition, a prototype was constructed. It appears that this system was successful technically (10).

2.5.3 Composite Semi-Elliptic (Leaf) Spring

A relatively new innovation in leaf springs is a fibreglass reinforced plastic (FRP) mono leaf spring (11). Although FRP springs have previously been used on Corvettes as a transverse leaf spring, the Budd Company has more recently developed a monoleaf which is of comparable size to a standard steel leaf spring and mounted in a similar longitudinal orientation yet half the weight. One application of this spring was on a Ford Econoline van in which a nearly 7% overall vehicle weight reduction was achieved as well as good ride quality. Another advantage to FRP leaf springs is in the orthotropic material properties of the spring. Because fibres can be oriented in specific directions, the failure mode of the spring (if it occurs) can be preselected.

2.6 STATE OF THE ART SUMMARY

One of the primary concerns for an intercity bus suspension is ride quality. Due to this, a system such as described above which uses a coil spring assisted by a pneumatic spring with a microprocessor control system would provide a soft ride on long highway trips, yet for tight corners and braking, the suspension would become more firm thus decreasing roll and pitch motions. In addition, using a MacPherson strut suspension may have benefits in terms of space and weight savings.

The hydro-pneumatic strut could also be applicable to bus suspensions, however, in order to compete with the air spring, some additional "active" features apart from levelling would have to be incorporated into the system, as compensation for the increased complexity.

Finally, FRP leaf springs, despite their light weight and good ride quality are not as applicable to modern buses since they are passive elements.

During the course of this review, it became apparent that little quantitative data is available for ride quality and handling of intercity buses. This creates difficulty in assessing the relative ride and handling performance between buses, and judgements must be made based on subjective sources and on the technical merits of each suspension layout. A grading of the buses is not included in this report due to the subjective nature of the performance information.

Ride on North American coaches is considered adequate, yet significant improvements could be made, particularly for travel on rough road and for long duration journeys. In addition, improvements in road handling of North American coaches would be desirable and would also contribute to a more pleasant ride. These conclusions are necessarily qualitative in nature and are based on the collective experience of UTDCI and its subcontractors who have ridden on North American and European coaches.

In contrast, some European coaches exhibit better ride quality and handling characteristics. This is due, in part, to the increased number of European suspension designs in existence. More testing has been carried out and design improvements have evolved at a more rapid rate. An additional factor is the wider public acceptance of intercity coaches in Europe for charter type services. This leads to initiatives for design improvements especially related to ride comfort.

It will be mandatory, however, that any suspension design improvements must take into account the harsher road environment in North America and the requirement for rugged, low maintenance equipment.

Independent suspensions, in theory, should provide a "better" suspension. These advantages contribute to both better handling and better ride. Despite the theoretical supremacy, independent suspensions (front) are still in many circumstances inferior to beam axle suspensions. One apparent reason for this is that beam axle suspensions are currently very "well engineered" in comparison to independent bus suspensions. Until recently, most independent suspensions have adapted standard components (such as wheel hubs) from beam axles and applied them directly to independent suspension designs, with resulting compromises in the geometry and hence performance. However, a properly designed independent suspension station should enhance ride and handling, and offer obvious packaging advantages.

"Adaptive Control" is a suspension system design trend from the automotive industry which offers potential benefits to the intercity bus. Most buses already possess a self-levelling capability (a form of adaptive suspension) to overcome problems of high laden to unladen weight.

To provide better ride over long journeys, an adaptive suspension could change its stiffness characteristics producing a very "soft" and comfortable ride, when the feedback mechanisms sense this type of driving. However, on shorter duration rides, in which more manoeuvering may be required, the suspension could operate in a "firm mode" thus giving better handling.

Implementing an "active" suspension to the highway bus would cause the suspension parameters to continuously change to inputs whereas the adaptive system would only "adapt" as a result of persistent global variable changes. Active suspension is dealt with in more detail in Section 7.0.

3.0 COMPUTER SIMULATION

3.1 TYPES OF SUSPENSION SIMULATIONS

The advantages of a good simulation stem primarily from the close approximation to reality which becomes available to the designer without the need for expensive prototype vehicles. In essence, experiments can be carried out on the computer with the results of parametric variations being studied almost immediately after the decision to try a new configuration is reached. These statements are of course only true for simulations which are fast and reliable. Speed comes from a wise choice of simulation type and reliability comes from the same choice coupled with experimental verification of the simulation. This section details the available choices. Subsequent sections narrow the list to include only those simulations which are capable of providing the level of realism expected by a designer.

3.1.1 Kinematic Simulations

The simplest and most common analysis performed on suspension systems is kinematic, wherein all links are assumed to be rigid, all joints are assumed to be perfect pins or sliders, and calculations are often performed graphically or with a hand calculator. The analysis may account for angular displacements as being small (linear kinematic analysis) or large (non-linear analysis where sines and cosines of angles are significant). The second of these can only be handled in the general case if a digital computer simulation is used.

Simulations of this type are useful during the initial design phase when the lengths of links and locations of pins are being decided. Their utility is limited however by the inability to include inertial and elastic effects and to account for the forces acting within the system.

3.1.2 Elasto-Static Simulations

An elasto-static analysis is very similar to a kinematic analysis in that inertial effects are excluded from the outset. The result is that particularly important dynamic design information such as natural frequencies, damping ratios, and response amplitudes cannot be determined. On the other hand, the elasto-static analysis allows for deflections of links under load, linear or non-linear geometry, and accounts for the static force distribution within the links comprising the system, thus surpassing the capabilities of a strictly kinematic analysis.

A common example of an elasto-static analysis is that performed on a structure in order to calculate its deflections in response to a prescribed set of applied loads. This is usually done by formulating a stiffness matrix for the structure and solving a set of simultaneous linear algebraic equations. A suspension system can be viewed as a complex set of interconnected links and elastic elements for which a stiffness matrix may be derived and used to find the deflections and forces acting within the components. This approach is linear by nature and cannot account for geometric non-linearities. It is common however, to experience relatively small motions within bus suspensions so that a linear elasto-static analysis may provide significant insight into the behaviour, particularly the force distribution characteristics, of the system.

3.1.3 Dynamic Simulations

Dynamic simulations take many forms. They include all of the kinematic and elasto-static properties of the suspension system as well as the inertial and damping properties. A set of dynamic force and moment balance equations are derived to represent the behaviour of the system. These are, in their most general form, a set of simultaneous non-linear second-order differential equations as opposed to the algebraic equations found in the previous sections.

The nature of dynamic simulations leads to increased realism since no physical effect, no matter how non-linear or otherwise difficult to model, need be removed from the analysis. This type of simulation also leads to increased cost in terms of human effort required to derive governing equations and computer effort required to effect a solution to these equations. Many forms of solution become available to the analyst as well. For example, a linearized version of the equations of motion can be used in the frequency-domain to study stability of equilibrium states of motion or to predict the frequency response characteristics of the vehicle. The fully non-linear form of the equations can only be used to determine a time-domain solution in which response variables are predicted as functions of time and may be plotted as they would appear from the recording of an experimental trace. The latter approach is the most time consuming in all respects but is the only approach which preserves the realism of the simulation since few simplifying assumptions need to be made.

A feature of dynamic vehicle simulations which must be addressed is related to the large and relatively flexible body which is supported by the suspension system. The analysis must include the bus body as either an elastic structure having its own natural frequencies and mode shapes or as a single rigid body which responds to the input forces without deflection, serving only to transmit forces from one point of interconnection with a suspension element to another. The second approach is commonly used in vehicle dynamics and is valid as long as the natural frequencies of the body are well above those of the rigid body on the suspension system. If the frequencies are not well separated, extra information about the elastic behaviour of the body is required. These details can only be gathered from extensive finite-element analysis or modal-analysis testing.

3.2 SIMULATION SPECIFICATION

This section details the requirements of the simulation in terms of satisfaction of project objectives. Consideration is given to all criteria whether of primary or secondary importance.

3.2.1 Project Objectives

Broadly stated, the overall objectives of the project are to improve and generalize the suspension/vehicle interface model. The intent is to develop a design tool which can be relied upon to provide accurate and detailed information about the performance of bus suspension systems. Specifically, the design tool is to take the form of a computer package which may be used effectively by a person skilled at suspension design but perhaps not as skilled as a computer programmer or operator.

3.2.2 Characteristics of the Design Tool

To be successful as a design tool, the computer package must:

- provide a verified, realistic simulation of bus dynamic performance;
- include all significant suspension elements, without simplification or assumption if at all possible.
- provide flexibility in terms of input definition so that the user mav easily change parameters and obtain results for the new configuration.
- provide meaningful output so that the user may easily interpret predicted performance.

3.2.3 Basic Requirements of the Simulation

In addition to the overall design tool characteristics stated above, there are certain specific requirements which must be met by this particular simulation.

The simulation must be able to provide the user with detailed predictions of:

- accelerations within the vehicle body;
- deflections and forces acting in important suspension elements;
- yaw rate of the vehicle; and
- lateral acceleration of the centre of mass.

The simulation must allow the user to:

- simulate very different vehicles without deriving equations or writing computer code;
- focus on vehicle suspension subsystems (e.g., front suspension only) without requiring a change in the input description of other parts of the vehicle; and
- model the details of a suspension system without resorting to "lumped" or "effective" models at the outset.

In meeting these requirements, the simulation must provide, as output, measures of the vehicle performance. These measures, chosen to reflect the concerns of suspension designers, are to include ride quality, as determined from the acceleration response to road irregularities experienced by passengers within the vehicle, and handling characteristics, defined as the response of the bus to prescribed driver steering inputs at speed.

3.2.4 Desirable Attributes of the Simulation

There are characteristics which are of value to the simulation but cannot be seen as essential. They are:

- ease of input descriptions of the vehicle, perhaps using menu type prompts;
- low cost, as reflected by a low initial cost for software and subsequent low costs for maintenance and operation; and
- the ability to run the simulation on small computers, thereby avoiding the necessity of using a main-frame computer.

3.3 SELECTION OF A SIMULATION OPTION

The requirements placed on the simulation in the previous section rule out the use of other than a dynamic simulation package. The need for predictions of inherently dynamic variables such as acceleration and yaw rate dictates that strictly kinematic or elasto-static simulations be dropped as candidates for the simulation package. The utility of these analyses must however be kept in mind. An approach that allows this type of information to be extracted from a dynamic simulation is desirable.

The three approaches available for development of the dynamic simulation can be summarized as follows:

1) A detailed derivation of the equations of motion for a typical bus configuration, followed by the production of a special purpose computer code to solve these equations.

2) The use of a dynamic analysis package developed for bus simulations by another organization. This option would include the use of modified versions of programs meant to simulate other types of vehicles.

3) The use of a generalized dynamic simulation package capable of automatically deriving and solving the equations of motion for arbitrary sets of rigid bodies. There is a necessity to modify these programs to suit the input forces peculiar to road vehicles.

To aid with the selection of an approach, contact was established with researchers at the National Research Council (NRC), Concordia University, the University of Michigan Transportation Research

Institute (UMTRI), Mechanical Dynamics Incorporated (MDI), and Queen's University. Persons at these establishments, known for their work in the field of vehicle dynamics, were queried in order to determine the availability of software packages suitable for this project. Available literature was also surveyed in an effort to locate references to suitable simulations.

3.3.1 Available Options

The result of the investigations was a short list of candidate simulation packages fitting into options 2 and 3 above. Since it was determined that some software already existed, the option of developing a totally new package specific to buses and with the governing equations derived by hand was eliminated. The remaining options were:

1) A set of dynamic simulation programs developed by UMTRI for heavy truck/trailer configurations. These simulations are widely used and validated. They are available in the public domain and versions are running at NRC and at Concordia. The vehicle models are of the lumped parameter type wherein suspension elements are considered in groups and have their properties lumped together as a single element. These simulations are meant for handling performance only, ride quality output not being predicted. The input of suspension properties is tailored to existing vehicles where large-scale static tests can determine the overall behaviour of an installed suspension. It is not possible to simulate complex suspension systems in a detailed fashion using the UMTRI programs.

2) The use of one of the programs now available for automatic derivation and solution of equations governing three-dimensional, non-linear, multi-body dynamic systems. Three such packages were considered, the first being the ADAMS package developed by MDI. This package is gaining favour in the United States for its abilities to handle simulations of a complexity and level of detail which could not be considered if equations were derived by hand. The package has a high purchase price, roughly $40,000 US, and requires quite a high level of training in order to operate it efficiently. ADAMS has been used in handling performance predictions of highway vehicles with good results.

   The other two programs considered were MEDYNA, developed in Germany by DFVLR, and RESTRI, a product of Queen's University. The first of these has a purchase price equivalent to that of ADAMS. It has been used primarily for rail vehicle simulations where flexible guideways were of importance making it the only package which currently has the ability to simulate elastic body modes. It requires large scale computing hardware and a high level of operator training. RESTRI is a similar program which is still in a development state. It has been shown to provide good simulations of simple rail vehicles and other dynamic systems. It is able to run on small computer systems but requires a significant level of effort on the part of users before they feel comfortable using it.

3) The use of a package which automatically derives equations of motion which have linear geometry (small angle approximations) but include non-linear component properties. The package A'GEM, developed at Queen's University, is highly interactive and capable of simulating large systems on small computers. The solution of the governing equations, once derived, must be handled by another program specifically designed for the simulation at hand. UTDCI has software capable of solving these equations with relatively minor modifications. The initial cost of the A'GEM program is $1200 for source code and significantly less for a microcomputer package.

3.3.2 The Selection

Based on the above information and the requirements of the design tool, it was decided that the most promising approach involves the use of the A'GEM package coupled with special purpose solution routines available at UTDCI. The use of the UMTRI software was ruled out because it could not provide the level of suspension detail required by a designer and could not simulate ride quality of a vehicle. The alternative of using a fully non-linear simulation, probably ADAMS if this were the decision, is attractive for reasons of realism in the simulation but was ruled out on the basis of cost of purchase, level of training required of the user, and on-going costs for maintenance and hardware.

The package which will be used for the bus simulation will therefore take the form of a series of modules, each of which will be easily used by the designer and which will provide a variety of output variables in line with those required. The model will include non-linear suspension elements thereby limiting the solution technique to the time-domain. The ride quality results which are typically quoted in the frequency-domain will be obtained by post-processing the time varying output accelerations as would be done for experimental results. The handling performance results are interpreted in the time-domain and will require no post-processing. The software necessary to effect the solution and the post-processing exists at UTDCI and will be modified to fit into the design package.

4.0 TEST PROGRAM

The test program is described in detail in Attachment A-1 and summarized in this section.

4.1 TEST OBJECTIVES

The fundamental purpose of the test program was to provide data for the development and verification of a general suspension system computer simulation for the highway bus. Particular objectives established at the outset of the test program were to:

- instrument and test a modern highway bus for ride and handling performance; gather baseline data on the present performance of highway bus suspension systems for a range of operating conditions.
- measure vehicle and suspension component characteristics required as input to the computer simulation.
- modify the vehicle to known off-design conditions and re-test; gather sufficient performance data for comparison with computer predictions derived with corresponding conditions imposed.

4.2 METHOD OF APPROACH

The test approach focused on the principal objective to provide validation for the computer suspension simulation. To place this objective in context, the simulation is intended as a design tool at the component level; that is, the designer inputs the geometric locations and connection points of each suspension component, it's force/deflection/velocity characteristics, and the inertial properties (centre of gravity, moments of inertia) of each rigid body being connected. For specified road and steering inputs, the simulation will predict the ride and handling performance of the vehicle system.

The test program was planned to examine the suspension of a particular highway bus which would, in parallel, be mathematically modelled for the computer simulation. Component locations, properties, etc. were measured to substantiate the simulation input, and the bus was instrumented to measure the ride and handling performance and to offer check points for the computer simulation.

During the performance testing period, practical parametric changes were made to the actual vehicle and suspension system (off-design conditions). Thus, comparison could be made between the actual performance results and those predicted when corresponding changes were made in the computer model. In this manner, a high level of confidence could be reached in the general computer simulation of bus dynamic performance for use with advanced suspension concepts.

To carry out the tests, a modern highway bus was leased from a prominent bus manufacturer, Motor Coach Industries, for a period of three months from April 17 to July 16, 1986. The chosen bus, Model 102-A3, is representative of the contemporary highway bus industry in North America.

The particular bus, shown in Figure 7, was one of the first of the Model series, built in 1985 and used as an engineering prototype vehicle. It was identical to the production bus, but as yet had not been introduced into revenue service. Thus, somewhat more flexibility was allowed in the types of tests which could be carried out.

General characteristics of the bus are:

Length	12.2 m (40')
Width	2.6 m (102")
Height	3.4 m (134")
Gross Vehicle Weight	17,146 kg (37,800 lb)
Number of axles	3
Type of Axles	front, beam axle drive, beam axle, dual tires tags, independent
Type of Suspension	air springs, all axles

To meet the objectives of the program, a test matrix was produced, as given in Table 1, encompassing tests of the bus in its present configuration, and tests in well defined, off-design conditions.

The off-design tests were, of course, limited to practical, non-destructive changes such that the bus was returned in its original condition.

The various bus configurations are briefly described below.

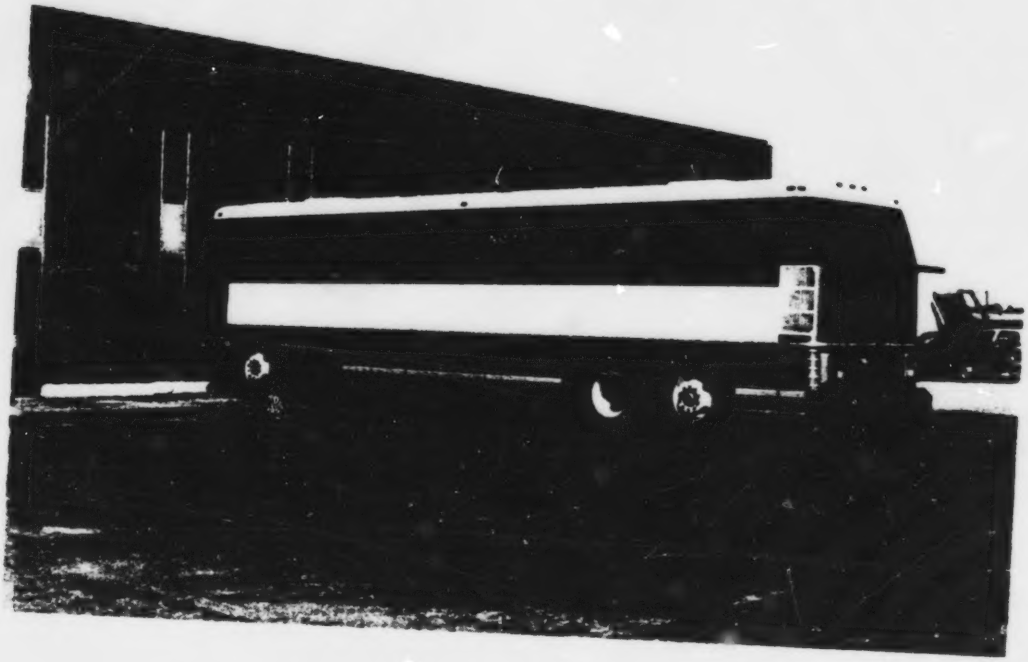


FIGURE 7: MCI 102-A3 TEST BUS

Test / Configuration #	Static	Dynamic					
		Ride Quality			Steady State Handling	Transient Handling	
	Weight/ c.g.	Random Input	Step Input	Harmonic Input	Constant Yaw Rate Input	Random Steer Input	Step Input
1							
2	X						
3		X	X		X	X	X
4						X	X
5		X		X			
6				X			
7				X			
8	X	X			X	X	X
9					X	X	X
10	X					X	X
11	X				X	X	X
12		X					
13		X					
14		X					
15		X					
16		X					

TABLE 1: TEST MATRIX

CONFIGURATION #	DETAILS
1	Bus unloaded, standard tires (pressures 111 psi-front, 75 psi-drive, 65 psi-tag) (Condition as received from bus manufacturer), tags down.
2	Bus unloaded, standard tires (pressures 111 psi-front, 75 psi-drive, 65 psi-tag), tags chained-up.
3	Bus unloaded, standard tires (pressures 100 psi-front, 80 psi-drive, 70 psi-tag), tags down.
4	Bus unloaded, standard tires (pressures 100 psi-front, 80 psi-drive, 70 psi-tag), tags depressurized (not lifted).
5	Bus uniformly loaded, standard tires (pressures 100 psi-front, 80 psi-drive, 70 psi-tag), tags down.
6	Bus uniformly loaded, standard tires (pressures 100 psi-front, 80 psi-drive, 70 psi-tag), tags depressurized.
7	Bus uniformly loaded, standard tires (pressures 100 psi-front, 80 psi-drive, 70 psi-tag), tags depressurized, anti-roll bar disconnected.
8	Bus uniformly loaded, low profile tires (pressures - all at 100 psi), tags down.
9	Bus uniformly loaded, low profile tires (pressures - all at 100 psi), tags chained-up.
10	Weights shifted to front and back, low profile tires (pressure - all at 100 psi), tags down.
11	Weights shifted to front and back, low profile tires (pressures-all at 100 psi), tags chained-up.
12	Bus unloaded, low profile tires (pressures-all at 100 psi), tags chained-up.
13	Bus unloaded, low profile tires (pressures-all at 100 psi), tags chained up, drive shocks disconnected.
14	Bus unloaded, low profile tires (pressures-all at 100 psi), tags chained up, anti-roll bar disconnected.
15	Bus unloaded, low profile tires (pressures-all at 100 psi), tags pressurized, anti-roll bar disconnected.

CONFIGURATION #	DETAILS
16	Bus unloaded, low profile tires (pressures-all at 70 psi), tags pressurized.

A more detailed explanation of configurations is given in Attachment A-1.

Note: Standard Tires = Michelin XZA 12R22.5
Low Profile Tires = Michelin Pilote XZA-1 275/80R

4.2.1 Baseline Tests

The baseline configuration is essentially defined by configuration #3; the bus was unloaded (the curb condition) with standard bus tires (Michelin XZA 12R22.5) having pressures set to 100 psi-front axle, 80 psi-drive axle, and 70 psi-tag axle. The air springs for the tag axles were fully pressurized (35 psig).

In the baseline condition, the following tests were performed:

1) Weight and centre of gravity location. These tests are described in Section 4.3. (Note: Weight and centre of gravity location were actually measured with the bus in Configuration #2; since neither tire pressures nor chaining of the tags have a significant effect upon the measurements, this is considered a baseline test.)

2) Ride Quality - Random Road Input.

   Random road input consisted of driving the bus along pre-selected portions of highway at different speeds and recording all relevant ride data as detailed in Section 5.0.

3) Steady-State Handling

   Steady State Handling tests were performed by driving on a constant radius circle at varying speeds and recording all relevant handling data. Section 4.6 describes the Steady State Handling tests.

4) Transient Handling

   Transient Handling tests were composed of two types of steering inputs. The first was an input in the frequency domain called "Random Steer Input" which consisted of driving the bus at constant speed while steering left and right in a sinusoidal fashion (amplitude ±90° on the hand wheel) starting first at low frequencies (0.2 Hz) and then increasing gradually to higher frequencies (up to 2 Hz where possible). The intent was to achieve a reasonable steering frequency content over the range 0.2-2 Hz.

The second transient handling test incorporated a steering step input. The focus for this test was to measure a time domain response of the bus when subjected to a rapid steering step input. The technique here was to drive the bus at constant throttle in a straight line and, at the selected time, turn the handwheel, as rapidly as possible, to a particular position and hold there for a minimum of 3 seconds to allow a new steady state to be achieved.

4.2.2 Parametric Tests

The parametric tests involved the altering and removal of certain suspension components and repeating most of the previous static and dynamic tests. The parametric changes included the following:

- Depressurizing tag axle air springs:
- Chaining-up tag wheels;
- Changing tire pressures;
- Changing tire type;
- Disconnecting anti-roll bar;
- Disconnecting shock absorbers;
- Adding weight; and
- Shifting weight.

One additional test, performed with the bus in several of the "off-baseline" configurations, was the "harmonic input" test.

The "harmonic input" test consisted of placing the bus on a four post "shaker", (2 tag wheels and 2 drive wheels were placed on the four posts) and cycling the posts over sinusoidal displacements. The frequencies and amplitudes of the motion of all four posts was the same; however, the phase relationship between posts was either in-phase or 180° out-of-phase, side-to-side. The frequency range covered from 0.1 Hz up to 11 Hz with amplitudes of ±1/8", ±1/4" and ±1/2".

4.3 VEHICLE INERTIAL PROPERTIES

4.3.1 Measurements

The total vehicle weight and the overall centre of gravity locations were determined by supporting the vehicle at four axle points (tags chained up). Clearances between each of the drive tires and the front tires were measured to ensure the vehicle was level. Load cell readings were then made. Also, positions of the four load cells were noted.

The sprung mass plus tag axles was measured with four load cells under four body points. The body was raised approximately 20 mm by operating the air levelling valves manually. With the body in this raised position, the load cells were positioned and "shimmed" such that, as air was slowly exhausted from the air springs, all body points made contact with the load cells at approximately the same time. Finally, all air was exhausted from the drive and front air springs (note: tags were chained up) such that the body was supported entirely by the load cells, and the axles (unsprung mass) were supported by the ground. Load cell readings were made.

The mass of the sprung mass plus tag axles and front axle was then measured. In this case, two load cells were placed under the front axle, and two load cells supported the body near the drive axle. This time, air was exhausted only from the drive axle air springs such that the weight of the drive axle was supported by the ground. Again tags were chained up. Finally, load cell readings were made.

The mass of the sprung mass plus front axle was measured next. The procedure was exactly the same as with the previous case (sprung mass plus tags plus front axle) except that the tag axles were unchained but depressurized such that their weight was supported by the ground.

Measurements of the sprung mass (bus body) and overall centre of gravity locations were performed simultaneously with the previous mass measurement tests. The only additional data required was the location of the loadcells.

To measure mass moments of inertia, the bus was suspended by two overhead cranes; the front and drive tires resting in wheel cradles. Essentially, there were three modes of oscillation in which the bus could swing:

1) Roll (side to side swinging);
2) Pitch (forward-backward swinging); and,
3) Yaw (rotational motion about a vertical axis).

The objective was to "excite" any one of these modes pushing the bus manually such that a period of oscillation for that mode could be determined.

4.3.2 Results

The following masses were determined for the various components:

Vehicle Curb Weight	13,780 kg
Body Weight (sprung mass)	10,652 kg
Front Axle	916 kg
Drive Axle	1,806 kg
Tag Axles (combined weight)	406 kg

The body centre of gravity location in the horizontal plan was found to be:

500 mm behind the body of geometric centre
22 mm to the right of the body geometric centre.

4.4 SUSPENSION COMPONENT TESTS

The test objectives were to test static and dynamic properties of selected components from the Test Coach so that sufficient information was available to accurately model and substantiate the components' behaviour.

Specifically, the following components and properties were examined:

- Shock Absorbers - damping properties.
- Radius Rod - radial spring rates of the rubber bushings.
- Air Spring - spring rates as well as interconnection effects.
- Tires - non-rolling vertical spring rates.

The approach taken was to remove key components from the bus suspension system. Generally, the component being studied was installed in a material test machine at the Royal Military College, Kingston and tested dynamically for its force-deflection/velocity characteristics.

An appropriate amplitude of oscillation was determined for each component, which was then cycled manually at that amplitude quite slowly in order to view quasi-static characteristics. Next, the component was cycled sinusoidally starting at very low frequencies (0.1 Hz) and gradually increased until higher frequencies were reached (2.0 Hz - depending on the amplitude). At higher frequencies, the amplitudes were reduced since the test machine used was limited in travel speed. (For details see A-1) A drive axle shock absorber is shown under test in Figure 8.

- Shock Absorber

  The shock absorber force was found to be primarily a function of relative velocity between the two ends of the shock and temperature for any shock absorber. Also, each shock absorber behaved very differently in compression than extension. In compression, forces through the shock were quite small and tended to vary linearly with velocity; as well the temperature effects in compression did not appear too significant. In extensions, the forces were much larger and were more non-linear with velocity (usually a second order polynomial fit worked quite well). Temperature effects in extension were quite significant (30% change in force for a temperature change from 12°C to 28°C); however, the changes occurred most rapidly in the temperature range of 12°C to 16°C. Coloumb friction in all dampers was very similar and was found to be approximately 180 N in both extension and compression.

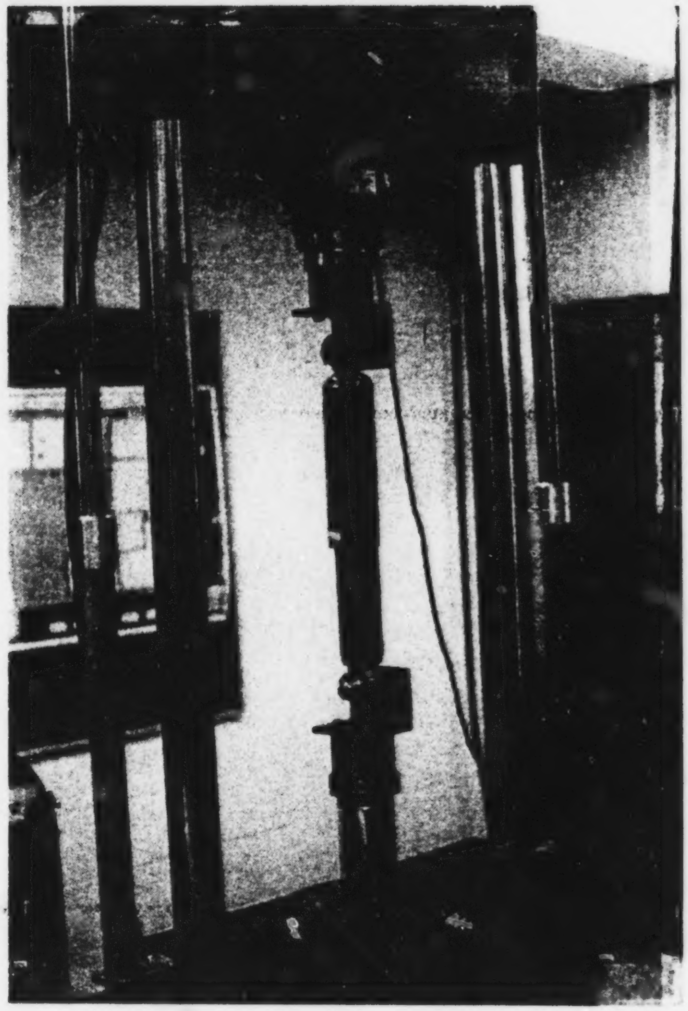


FIGURE 8. DRIVE AXLE SHOCK UNDER TEST

- Radius Rod

The radius rod data produced a static stiffness for the rubber bushings of 77,000 lb/in. The value quoted by the manufacturer was 75,000 lb/in. A dynamic spring rate was also quoted by the manufacturer of 1.5 times the static rate. Unfortunately, the MTS machine was not capable of achieving these results; however, a slight increase of approximately 5% the static stiffness was noted for higher frequencies.

- Air Spring

Without a reservoir connected, the air bag stiffness was found to be 1,130 lb/in and constant over the frequency range 0 - 5 Hz for the conditions given in Figure 9. With the reservoir connected, the stiffness varied from 750 lb/in statically to 1,240 lb/in for high frequencies.

- Tire Data

From Figure 10, the compression of a tire from zero load to maximum load is non-linear (again a second order polynomial fits the data well). However, over shorter portions of the curves (±1/4") and constant stiffness could be assumed.

The frequency of oscillation also had some effect on the stiffness of the tires. As frequency increased, the stiffness increased. It appears that the standard bus tires stiffness was more affected by oscillations than the low profile truck tire.

The vertical stiffnesses of the tires at 100 psi tire pressure were the following:

Low profile tires:	5,360 lb/in
Standard tires;	5,330 lb/in

4.5 RIDE QUALITY

The performance of the bus suspension system in providing good ride characteristics for the passengers was tested with the objective of measuring the response of the test bus body (ride quality) to vertical inputs to the bus tires for a range of operating conditions. An additional objective was to measure axle response as complementary information for the validation of the computer simulation. Ride quality instrumentation was set-up to measure

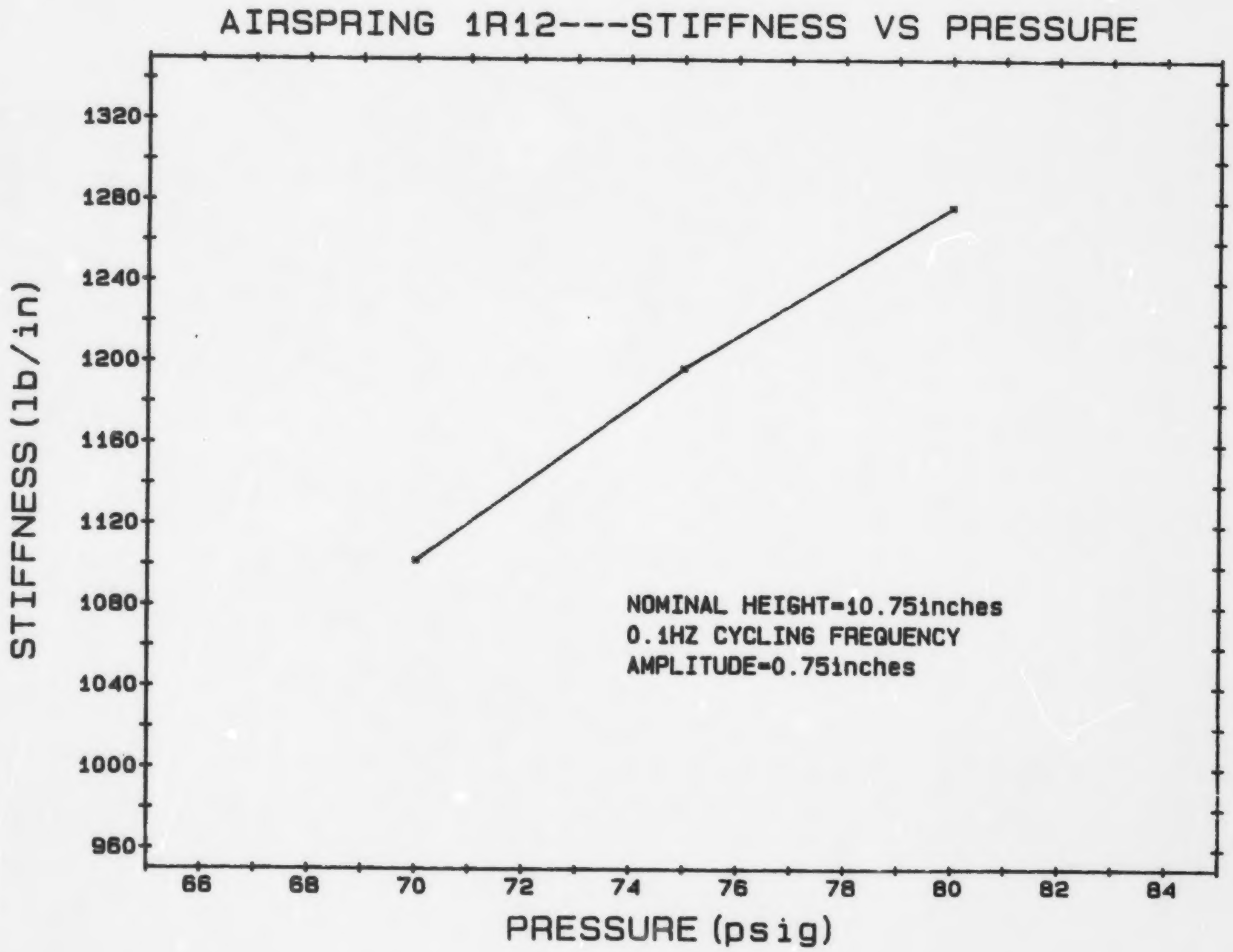


FIGURE 9: TYPICAL AIRSPRING STIFFNESS DATA

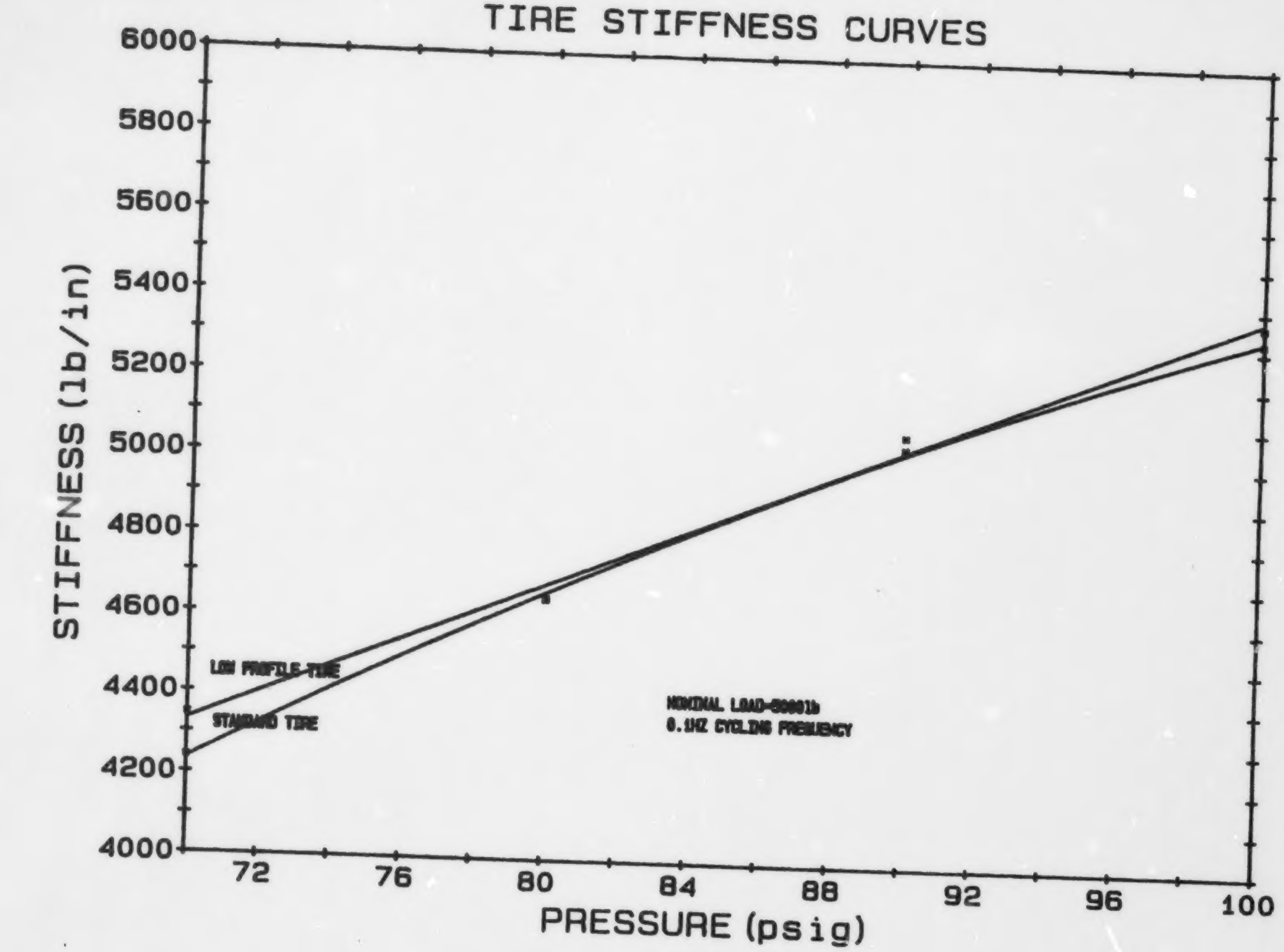


FIGURE 10: TYPICAL TIRE STIFFNESS DATA

accelerations within the bus body, accelerations on the axles, displacements of the axles relative to the body and forward speed of the bus. This instrumentation is summarized in Table 2. The vertical input at the bus tires was derived in three basic manners as given below.

4.5.1 Random Road Input

For Random Input, the bus was driven along various preselected portions of highway (of varying roughness grades) at different speeds, while the ride instrumentation measured the relevant parameters.

4.5.2 Step Input

The bus was driven over a railway crossing which was sufficiently rough to simulate a 'step' and thus excite the suspension natural frequencies. The ride instrumentation measured necessary parameters as the test bus was driven at different speeds over the railway crossing.

4.5.3 Well - Defined Harmonic Input

The bus was placed on a four post shaker and the tag and drive wheels were excited at different frequencies, amplitudes and phases while the ride instrumentation recorded all meaningful parameters.

4.5.4 Test Procedures

The procedures for performing a Ride Quality test were the same regardless of the input (Random, Step or Well-defined). In each case all power supplies were turned on and all necessary connections to the junction box were checked to confirm the correct instrumentation for Ride Quality. Next, the excitation voltages from the power supplies were checked to confirm the correct levels (one supply at ±10V and the other at ±15V). All tape channels were then calibrated by simultaneously recording standard DC signals (+3.77V, 0.0V and -3.72V) to all channels. This was done usually at the beginning and end of each ride quality test. Any additional relevant information during a test was recorded on the voice channel of the Sangamo recorder.

- Random Input

  For Random Input, the bus was driven along a road at a constant speed (cruise control usually engaged) starting at 20 mph going in steps of 10 mph until 50 mph was reached and then from 50 mph back to 20 mph.

TRANSDUCER TYPE	QUANTITY	DESCRIPTION
Accelerometers		
	3	±2g vertically oriented on bus body (front, middle and rear of bus)
	2	±1g laterally and longitudinally oriented (rear cg. of bus body)
	2	±5g vertically oriented (centred on each axle)
	1	±2g laterally oriented on bus body (rear body c.g.)
Displacement Transducers		
	4	±100 mm; Vertically oriented between axle and body (2 per axle)
	2	±100 mm; Laterally oriented between axle and body
	1	±100 mm; Between steering tie rod and front axle
Rotary Displacement Transducers		
	1	±950°; Measurement off hand wheel shaft
	2	±20°; Balanced Slip/Roll trolley
Speed Transducer		
	1	0-34 m/s; Operates off cruise control
Recording Instrumentation		
	1	14 Channel FM analog tape recorder
	1	Dual channel Data Acquisition Micro Computer with printer and CRT.
Miscellaneous		
	1	Voltage Inverter (24V - 115V)
	2	DC Power Supply
	1	Attenyator junction box (for transducer cables)
	1	Digital Volt Meter
	1	Digital Thermometer (for tire temperature)
	1	Air pressure gauge

TABLE 2. SUMMARY OF TEST INSTRUMENTATION

- Step Input

  The test bus was driven at constant speed over the Celanese Rail crossing at speeds of 20, 30, 40, and 50 mph while data from the transducers was recorded. Typical results are shown in Figure 11.

- Well-Defined Input

  With the bus drive and tag wheels on the four post shakers, various harmonic inputs were requested. (Communications was via remote headsets). Once the shaker posts began to operate, several seconds passed before recording started to allow transient vibrations to decay.

  Tests were carried out with the shakers both in and 180° out of phase for a wide variety of input frequencies from 0.5 to 15 Hz and several different amplitudes.

  Similar tests as above were run with tag axles depressurized and with the torsion bar disconnected.

  The harmonic tests were carried out with Test Bus configurations #5, 6 and 7. Typical results are discussed in section 6.0.

4.6 STEADY STATE HANDLING

One measure of vehicle handling performance can be determined through steady state handling tests. Steady state handling can be considered as the vehicle response to constant steer wheel input or to constant path.

The test approach followed procedures outlined by the International Organization for Standardization (I.S.O.), Report 4138, entitled "Road vehicles - Steady state circular test procedures".(12) The bus was driven around a constant radius circle at increments of increasing forward speed.

The vehicle steady-state response was measured through such variables as the road wheel angles, roll angle and slip angle of the bus. Also, such parameters as the forward velocity, lateral acceleration and handwheel angle were measured.

Slip angle and roll angle were measured using a small balanced trolley system attached to the rear of the coach.

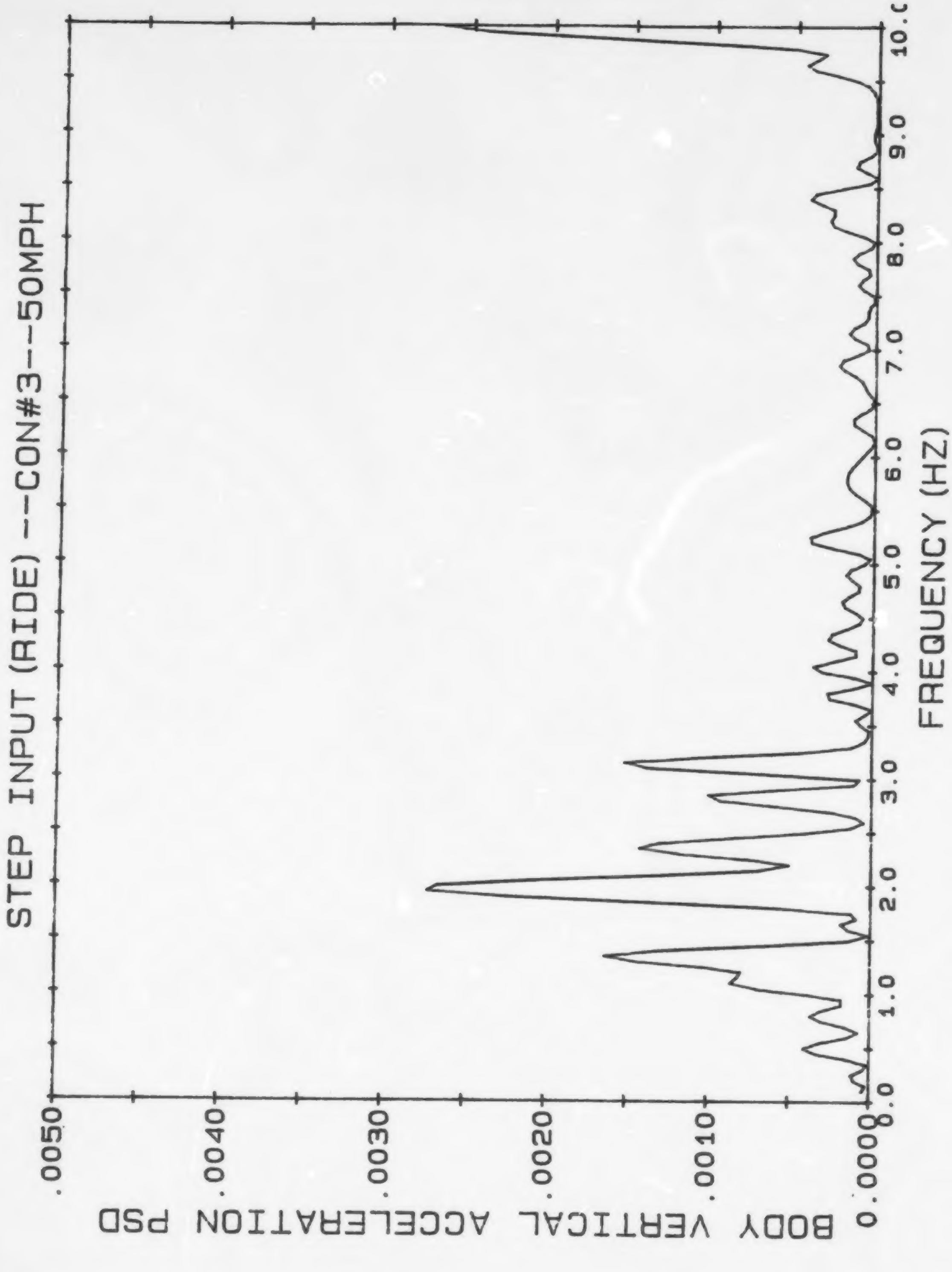


FIGURE 11: TYPICAL RIDE TEST RESULTS FOR STEP INPUT TO BUS

4.6.1 Test Set-Up and Location

Tests were carried out on uniform smooth, dry, concrete and tarmac surfaces with high friction levels (tire to surface 0.6). There was less than 2% gradient over the test areas.

The condition of the test tires, (important role in vehicle handling), was less than half worn with good shoulder condition. Care was taken that tire temperatures were not excessively high or low throughout the tests (30-45°C).

Site preparation involved painting a white circle on the paved surface. For this purpose, a mixture of lime and water was used, with a paint brush. A centre was first located, and a stretched length of nylon rope pivoted from this centre point to follow the circumference of the circle while painting.

Three separate tests were utilized for the steady-state handling tests. The first site was in the parking lot at UTDCI. A 15 m radius curve was formed, and used to familiarize personnel with test techniques. The second site was at Mountain View airport at the junction of the two runways where a 35 m radius circle was formed on the tarmac surface. The third site was at the Transport Canada Vehicle Test Site near Blainville, Quebec in an area of the test site known as "Zone Delta". In Zone Delta a 40 m radius circle was painted.

The test bus was set-up in the appropriate configuration before a test could begin. The fuel level was usually maintained around 3/4 full to prevent fuel spillage. The levelling valves were disconnected before the steady-state tests to ensure that the valves would not try to level the bus during the steady-state manoeuvers. Finally, instrumentation was connected as per the steady-state handling tests.

Results are shown for one of the tests in Figure 12.

4.7 TRANSIENT HANDLING

Another measure of vehicle handling performance is through tests of transient handling response.

The chosen tests followed procedures outlined by the International Organization for Standardization (I.S.O.), Draft Proposal 7401, "Road Vehicles - Lateral Transient Response Test Procedures", September 1984. (13)

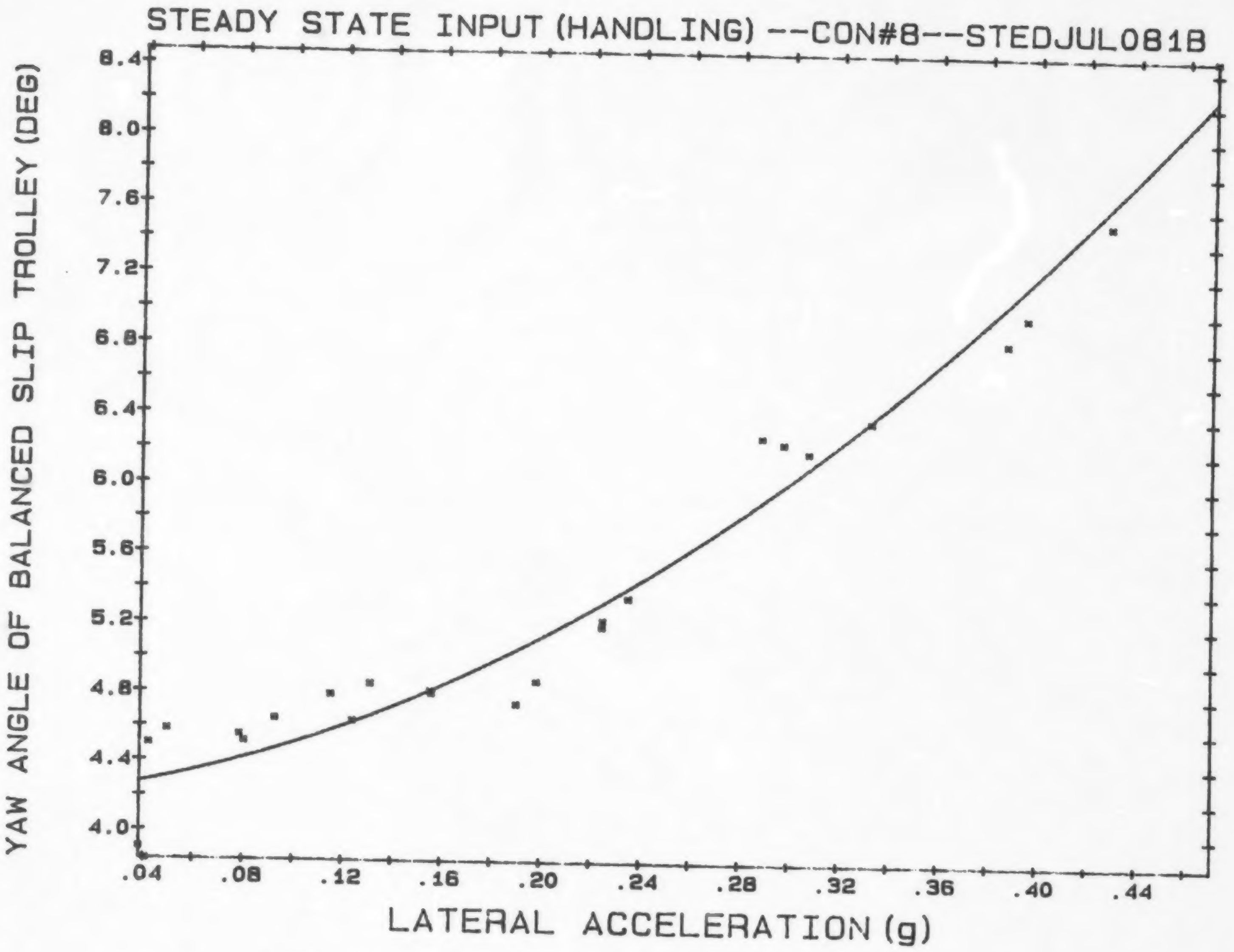


FIGURE 12: TEST RESULTS FOR STEADY STATE HANDLING

Transient handling test results can be divided into two categories: Those involving a time response and those involving a frequency response. For this test program, one test input was chosen from each of the two categories. From the time response category, a Step Input was selected. From the frequency response category, Random Input was selected.

For the case of a step input, the test bus was driven at a constant speed in a straight line, and at a selected point, a sudden steering input was induced causing the bus to turn sharply while maintaining throttle. This pseudo-step (ramp) input was held constant for several seconds allowing all transients to decay. A time history was recorded of the step input and the response of the test bus.

The Random Input test began with the bus driven at constant speed in a straight line. At some point, the driver began steering "erratically" by moving the handwheel clockwise and anti-clockwise in a sinusoidal motion. Initially the frequency was low (0.2 Hz) but was gradually increased to higher frequencies (1.5 Hz) trying to keep the handwheel turn amplitude approximately constant (±90°). During the run, the speed is held constant. When sufficient frequencies had been covered, or the driving space was becoming too short, the bus was again driven straight at constant speed to complete the test.

4.7.2 Test Set-Up and Location

Tests were carried out on uniform smooth, dry, concrete and tarmac surfaces with high friction levels (tire to surface 0.6). There was less than 2% gradient over the test areas.

The condition of the tires, (important in vehicle handling), was less than half worn with good shoulder condition. Care was taken that tire temperatures were not excessively high or low throughout the tests (30-45°C).

No site preparation was required as with the steady-state handling tests. However, before tests could be started it was ensured that there was adequate space to perform a test. This was usually accomplished by driving through a "mock" test.

Several test sites were used for performing transient handling tests. While practicing step inputs, the UTDCI parking lot was used. For practicing random inputs, Taylor-Kidd Boulevard near UTDCI was used. Actual transient handling tests were performed at Mountainview Airport (Belleville) and the Transport Canada Vehicle test site near Blainville, Quebec. At Blainville, step input handling tests were conducted in the dynamic test area known as "Zone Delta". Random input tests were conducted on the high speed test track known as "Zone Bravo".

Typical results for a step input are shown in Figure 13. Random input test results are discussed further in section 6.0.

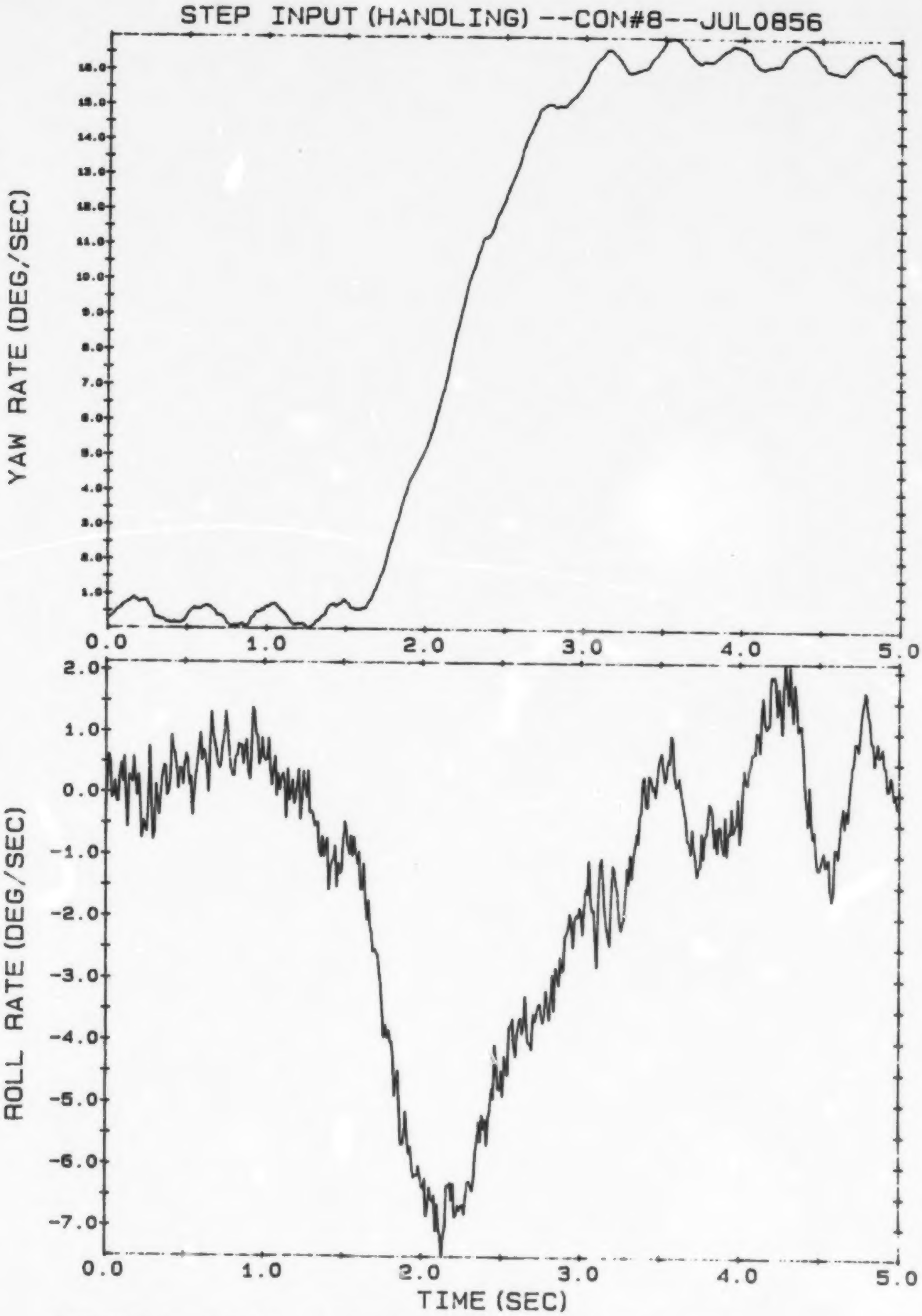


FIGURE 13: TYPICAL TRANSIENT HANDLING RESULTS FOR STEP INPUT

5.0 MODEL VERIFICATION

The computer simulation package was developed as a design tool, subject to the characteristics and specifications outlined in Section 3, to provide accurate, detailed information about the performance of bus suspension systems.

5.1 GENERAL APPROACH

The approach taken in the development and validation of the simulation package is best described as a series of discreet steps:

Step 1:

Computer simulation modules were developed to predict suspension performance. These are general computer programs. Performance is examined in 2 modules:

a) Ride performance - vertical input and response
b) Handling peformance - lateral input and response

Step 2:

A rigid body computer model of the Test Bus was created. The model is made up of many rigid bodies connected by suspension elements, and requires:

- Inertial properties of each rigid body component
- Force-deflection characteristics of each connecting element; e.g. springs, tires, shocks, bushings...

A combination of manufacturers' data, theory and component test was used to prepare the model.

Step 3:

The computer model of the Test Bus for a given configuration or set of conditions (e.g. curb condition, all components in revenue service condition, etc) was entered into the simulation modules. Suspension performance for ride and handling was predicted.

Step 4:

The suspension of the bus was tested in the given condition for vertical input (ride) and lateral input (steering). Data for suspension performance was analysed.

Step 5:

The test data was compared with the predicted performance data. A high level of confidence was assumed in the test data. The Test Bus Model (Step 2) was then updated using justifiable criteria and the simulation modules improved (Step 1) until agreement was reached.

Step 6:

The suspension of the bus was tested in other (off-design) configurations for vertical and lateral inputs. Data for suspension performance was again analysed.

Step 7:

The rigid body computer model of the Test Bus was adjusted (Step 2) to the given (off-design) condition or configuration used in the tests (Step 6). The simulation modules were re-run (Step 3) with the adjusted test bus model.

Step 8:

The test data was compared with the predicted performance data for each (off-design) configuration. The Test Bus Model (Step 2) and the Simulation Modules (Step 1) were re-examined as required. A consistency in the model and simulation modules was developed with the test data.

5.2 COMPUTER SIMULATION MODULES

The problem of providing a detailed dynamic performance simulation for arbitrary buses was addressed by using a general purpose program designed to provide computer-generated equations-of-motion for linear, three-dimensional multibody systems. The premise is that the majority of terms in the equations of motion for a complex vehicle model will arise from linear stiffness and damping elements within the physical system.

The method was then extended to allow for the inclusion of suspension elements having non-linear constitutive relationships. Since angular motions found within the bus suspension systems are generally only a few degrees in magnitude, non-linear geometrical terms arising from trigonometric relationships have not been included.

The simulation package was developed as a group of modules, each run separately with data transfer between modules accomplished using data files created and saved in disc storage. The program modules were designed and written to be implemented on a mini or micro computer system. A block diagram of the various modules is shown in Figure 14.

The principal modules are described below.

5.2.1 A'GEM Module: Automatic Generation of Equations of Motion

The A'GEM program was designed to simplify the process of obtaining linear equations-of-motion for three-dimensional multibody systems. Simple input files are required for this program in order to simulate the dynamics of arbitrary systems of rigid bodies without

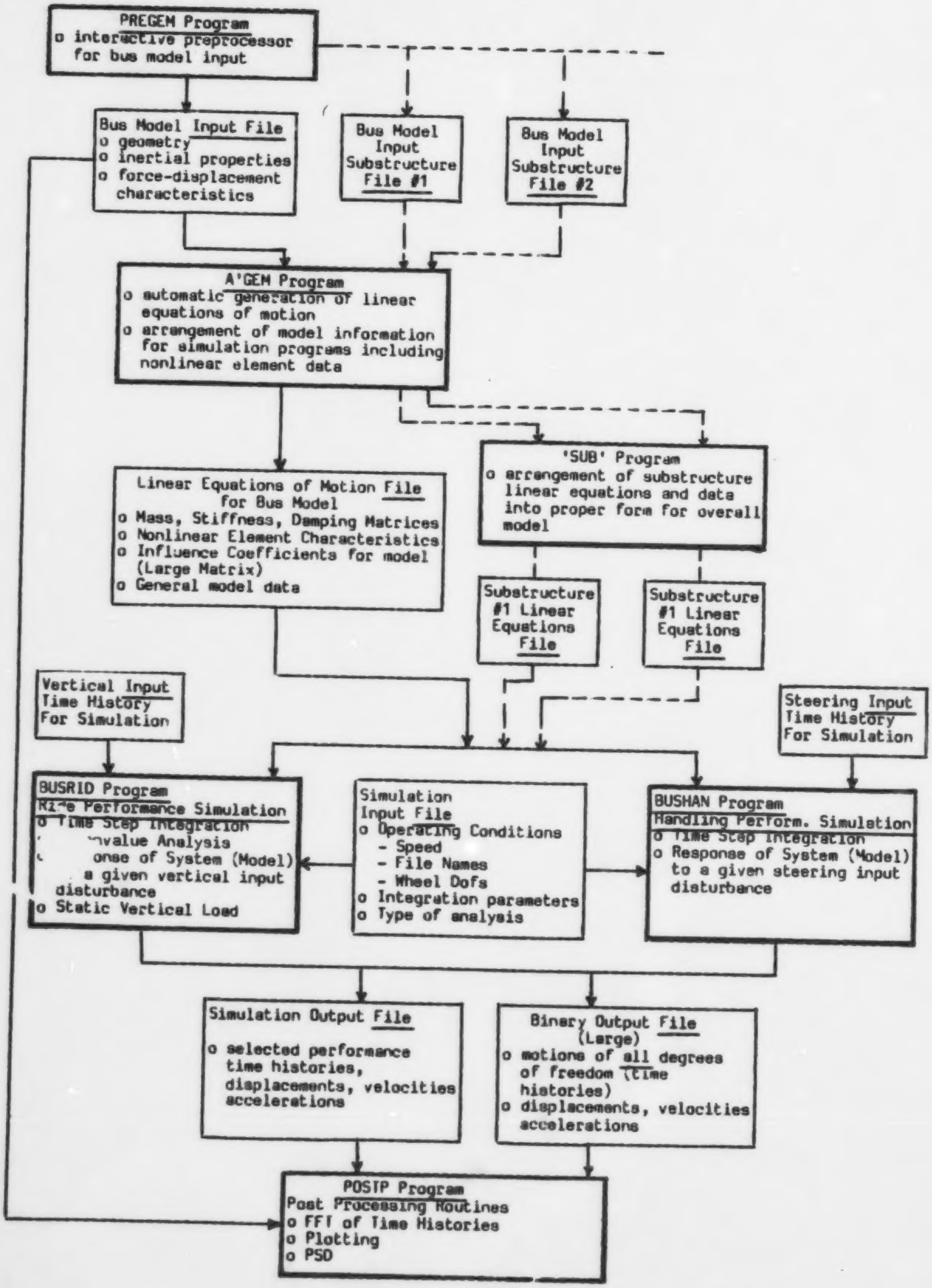

PREGEM Program
o interactive preprocessor for bus model input
Bus Model Input File
o geometry
o inertial properties
o force-displacement characteristics
Bus Model Input Substructure File #1
Bus Model Input Substructure File #2
A'GEM Program
o automatic generation of linear equations of motion
o arrangement of model information for simulation programs including nonlinear element data
'SUB' Program
o arrangement of substructure linear equations and data into proper form for overall model
Linear Equations of Motion File for Bus Model
o Mass, Stiffness, Damping Matrices
o Nonlinear Element Characteristics
o Influence Coefficients for model (Large Matrix)
o General model data
Substructure #1 Linear Equations File
Substructure #1 Linear Equations File
Vertical Input Time History For Simulation
Steering Input Time History For Simulation
BUSRID Program
Ride Performance Simulation
o Time Step Integration
nvalue Analysis
onse of System (Model)
a given vertical input disturbance
o Static Vertical Load
Simulation Input File
o Operating Conditions
- Speed
- File Names
- Wheel Dofs
o Integration parameters
o Type of analysis
BUSHAN Program
Handling Perform. Simulation
o Time Step Integration
o Response of System (Model) to a given steering input disturbance
Simulation Output File
o selected performance time histories, displacements, velocities accelerations
Binary Output File (Large)
o motions of all degrees of freedom (time histories)
o displacements, velocities accelerations
POSTP Program
Post Processing Routines
o FFT of Time Histories
o Plotting
o PSD

FIGURE 14: BLOCK DIAGRAM OF SIMULATION MODULES

any analytical effort. The theory embodied in this program can be found in Reference 14.

Briefly, A'GEM derives the governing equations-of-motion (mass, stiffness and damping matrices) by working directly with a description of the system contained in a simple input file. In addition, the program arranges data for non-linear suspension elements into a form suitable for simple inclusion into the forcing functions in particular simulation modules.

The physical system must first be modelled by the designer in conceptual form as a system of rigid bodies connected by suspension elements. Data must be gathered on the inertial properties of each rigid body, the geometrical location of each element and their interconnections, and the force-displacement-velocity characteristics of each interconnecting element. This information must be incorporated into an input file for A'GEM, and a preprocessor (PREGEM) has been written to assist in this function. The components of the conceptual model used by A'GEM are:

1. Rigid Bodies: these are bodies which have inertial and dimensional properties, and whose motions are the primary dynamic variables. The "Ground" is a special rigid body which is not allowed to move.

2. Reference Frame: the equations-of-motion are derived by allowing each rigid body to move in 3-dimensional space. The description of these motions requires a set of co-ordinate axes for reference, and a body may then have linear plus rotational motion about these axes.

3. The Origin: a point in space which is assigned co-ordinates (0, 0, 0). All positions are specified with respect to this point. The origin is not attached to any rigid body although it is often convenient to choose a location which coincides with an important point on a body such as a centre of mass.

4. Degrees-of-Freedom: the actual motions allowed for the rigid bodies. The user may allow each body to have freedom to move in any number of directions varying from a low of zero to a high of six. A body with no motion is, by definition, the ground. Other bodies may have no user specified degrees-of-freedom if their motion is completely governed by constraints. The degrees-of-freedom are numbered sequentially from 1 to the total number of degrees-of-freedom for the entire system.

5. Springs: these are the elastic elements which connect neighbouring rigid bodies. Each spring is characterized by its stiffness, the positions of its ends, and the bodies to which the ends are attached.

6. Dampers: dampers are analogous to springs with the only differences being that forces within them are proportional to the relative velocity of their end points. In A'GEM, it is assumed that every spring acts in parallel with a damper so that both stiffness and damping values are required for any element which is defined. This reduces the amount of positional data which must be provided. If an element consists of only a spring and does not produce damping forces, the input data simply specifies a damping coefficient of zero.

7. Non-linear Connection Elements: forces within these elements are non-linear functions of the displacement and velocity across the elements. Each element is characterized by the positions of its ends, the bodies to which the ends are attached and short tables of force versus displacement and velocity.

The mathematical model of the system is described by a set of simultaneous, linear, second-order differential equations, having the standard form:

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{F_{NL}\} + \{F\}$$

where M = mass matrix
C = damping matrix
K = stiffness matrix
$\{\ddot{x}\}, \{\dot{x}\}, \{x\}$ = vectors of acceleration, velocity and displacement of each degree of freedom in the system.
$\{F_{NL}\}$ = forcing vector containing the contribution of each non linear suspension element to each degree of freedom.
$\{F\}$ = forcing vector of external forces.

The above notation denotes one equation for each degree of freedom, for a total of "NDOF" equations.

The mass, stiffness and damping matrices are automatically produced by A'GEM. The matrices are square, having dimensions equal to the total number of degrees-of-freedom. The ordering of the equations is that specified by the user through the numbering of the degrees-of-freedom. These matrices have exactly the force which would arise from a derivation by hand.

The forcing vectors represent all forces of interaction of the vehicle with the environment as well as the forces acting within non-linear suspension components. Interaction forces with the environment are introduced in the particular simulation modules for ride and handling performance. A'GEM arranges the non-linear suspension component data into a simple form such that the forces arising from these components can be easily calculated at each time step of the particular simulation for inclusion into the forcing vector.

The inclusion of non-linear components is a straight forward extension to A'GEM, since the stiffness and damping matrices are actually generated by using influence coefficients. The effect of each degree-of-freedom on the deflection of each suspension element in the system is calculated from the connectivity and positional information supplied by the user. A rectangular influence coefficient matrix, (D), dimensioned NS (the number of suspension elements in the system) by NDOF (the total number of degrees-of-freedom) is formed. This information, together with the force-deflection-velocity data tables for each non-linear suspension component is passed to the specific simulation module (ride or handling) to facilitate the non-linear force calculations.

5.2.2 Ride Performance Module: BUSRID

Road vehicles are supported by rubber tires which interact with the road surface as the vehicle moves over it. The forces of interaction at the tire/road interface are responsible for all motion of the vehicle. Ride simulations are concerned with the vertical motions of the vehicle.

The highway bus rides on tires which follow the road surface vertically as the vehicle moves over it. In order to simulate the dynamic motions, the tires (on the contact patches between the tires and the road) must be included as individual bodies in the vehicle model before it is input into the A'GEM module. Dual tires may be considered as a single tire, in which case, the standard 3-axle highway bus generally requires 6 tire bodies. Each of these bodies must further be given a vertical degree of freedom for the ride simulation. In general, the tires are given longitudinal, lateral and vertical degrees-of-freedom in the vehicle model in order to accommodate tire inputs for the handling simulation as well (Section 5.2.3).

In BUSRID, the vertical displacements of each tire in time are used as the input to the simulation. That is, each tire vertical degree-of-freedom is constrained to move in its specified manner in time, and the equations of motion are integrated to determine the time response of the other degrees-of-freedom in the vehicle model to these inputs. General inputs include:

- the vertical "shaking" of any specified tire or tires at given amplitude and phase in a sinusoidal manner;
- distinct vertical inputs in time to each of the front tires which are followed in time (dependent on the vehicle speed and tire location) by the same vertical inputs to the succeeding tires.

The stiffness, damping and mass matrices for the linear equations of motion are simply passed to the simulation from the A'GEM Module output file. The user specifies which rigid bodies of the model are the tires and their input displacements with time. Initially, there is one equation of motion for each degree of freedom in the system.

The simulation automatically rearranges the equations, separating out (partitioning) the tire vertical degrees of freedom in order from front to rear of the bus and, for equal longitudinal location, from right to left side. These equations are manipulated algebraically and need not be integrated since these degrees-of-freedom already have specified displacements with time. The tires are grounded laterally in the simulation (constrained to not move laterally) by removing any tire lateral degrees-of-freedom from the system of equations. This reduces the number of equations to be solved and ensures a purely vertical input. It should be noted that vertical and lateral springs representing the tire stiffness are still included in the simulation model; the tire "body" is just a representation of the wheel/road interface.

The remaining equations may then be integrated to determine the time histories of all the other degrees of freedom in the vehicle model relative to the ground.

In BUSRID, before the integration is carried out, the user is given the option to make a preliminary examination of the vehicle model. With this option, the rigid bodies of the system are statically placed under the force of 1 g acceleration (gravity) in the vertical direction. The tires are constrained vertically to their ground positions, and the linear stiffness equations are solved to determine the displacements of the vehicle degrees-of-freedom. The system is thus easily checked for obvious modelling errors, and the static normal force through each tire is also found.

In addition, the linear equations are also solved to determine the characteristic frequencies (eigenvalues) and modes (eigenvectors) of the linear system. This generally assists in the overall examination of the suspension system.

5.2.3 Handling Performance Module: BUSHAN

The handling simulation module is concerned with the lateral motion of the bus resulting from the lateral forces of interaction between the tires and the road.

In order to write the equations-of-motion for the vehicle, it is necessary to derive expressions for the absolute (i.e. relative to ground) accelerations of the vehicle and to follow this with a force balance on each of the components. A reference body on the vehicle is chosen as the carbody itself (the largest mass) and is assigned the task of containing the basic degrees-of-freedom for the simulation. That is, this body has those motions to which all others are referenced. In Figure 15, the body is assigned a large forward velocity, U, which may or may not vary with time. The other basic degrees-of-freedom are the lateral velocity, V, and the yaw rate, r. The motion of this body is completely described in terms of these velocities. The position of the body is not defined nor required. It is a simple task to show that the acceleration of the body has an x component:

$$a_x = U - Vr$$

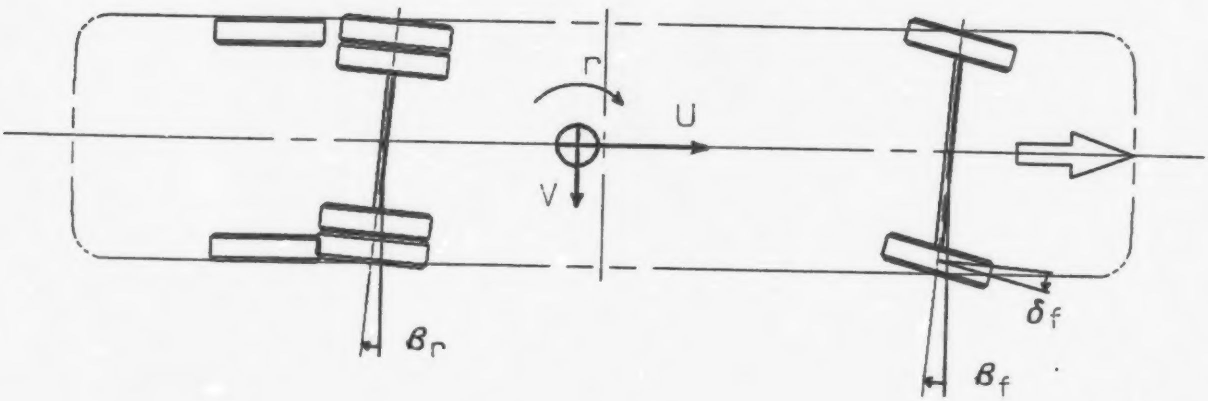
and a y component:

$$a_y = V + Ur$$

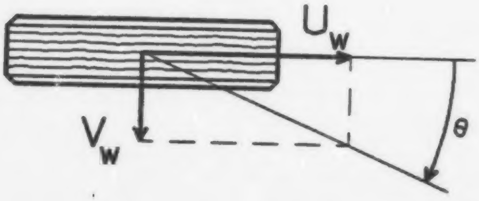
There is no "z" component to the acceleration because the assumption is made that the vehicle is travelling over smooth level pavement and that vertical motions will be small and therefore negligible. In fact, the reference body used consists of a horizontal plane passing through the centre of mass of the body. Using this definition allows the body to roll and pitch without loss of generality.

The acceleration of every other component of the bus model is found by applying a relative motion analysis. The remaining accelerations are defined to be equal to those of the reference body plus a term arising from the second time derivative of the position of the body in question with respect to the reference body. This definition is completely analogous to the definition used by A'GEM in deriving the linear equations-of-motion given in Section 5.2.1, except insofar as the reference body there was the ground whereas it is now one of the bodies in the system.

The most critical feature of the handling simulation is the description of the tire forces. These forces result from a misalignment between the direction a tire is headed and the direction it is moving (slip angle). The total velocity vector of the tire is therefore not aligned with the tire and a lateral force of interaction between the tire and the roadway is developed.


r
U
V
δ_f
β_r
β_f

U = forward velocity of body
V = lateral velocity of body
r = yaw rate of body
δ = wheel angle w.r.t. axle
β = axle angle w.r.t. body



U_W = forward velocity of tire
V_W = lateral velocity of tire
θ = slip angle of tire

FIGURE 15: REFERENCE BODY SHOWING DEFINITIONS OF DEGREES OF FREEDOM AND SLIP ANGLE

Experiments have shown that the lateral tire force varies with the slip angle in a manner shown in Figure 16 for the standard 12R22.5 bus tire. An empirical relationship developed by Calspan Corporation can be shown to produce a cubic variation where the coefficients of the cubic are functions of tire construction, tire pressure, normal load, and coefficient of friction amongst other things. The tire force saturates in the limit and becomes equal to a frictional force.

The bus handling simulation allows the user to specify particular tire force characteristics or to make use of a cubic variation of lateral force with slip angle which includes the saturation force levels. Since the slip angles are functions of the velocity vectors of the tire, it is necessary to calculate the tire velocities at every time step. Since the tire forces also depend on the normal load through the tire, this normal force must be computed as well at each time step. This is accomplished by ensuring that the bus model has each tire (or duals) specified as a body (as for the ride simulation) having a lateral degree-of-freedom. The velocity of the tire can then be calculated automatically as it is for any other body in the simulation. The resulting non-linear force is included as has already been described for forces in non-linear suspension elements.

This general approach to including tires in the model means that any number of tires can be simulated. Buses of any design fit easily into the structure of the simulation program.

The angle of the steered road wheels (tires) relative to the ground are used as the principal inputs to the handling simulation. The front steer angles are generally under driver control and are those usually specified. However, the capability is included to specify the wheel angles on any of the tires as a function of time.

5.3 BUS MODEL

A computer model of the 3-axle Test Bus was prepared to verify the simulation modules. The general dimensions and layout of this bus are shown in Figure 17, together with the type of suspension used with each axle.

The bus body is supported vertically at each axle by air springs, with a damper (shock absorber) in parallel with each spring. The front beam axle is positioned longitudinally with respect to the body by upper and lower trailing links, and laterally by a panhard rod. The beam drive axle is located by two lower leading links in the longitudinal direction and two upper links, diagonal so as to provide both lateral and longitudinal support. An anti-roll torsion bar is also connected between the bus body and the drive axle to provide additional roll stiffness. An independent tag axle (one each side), pivoting from the bus body about a support pin complete

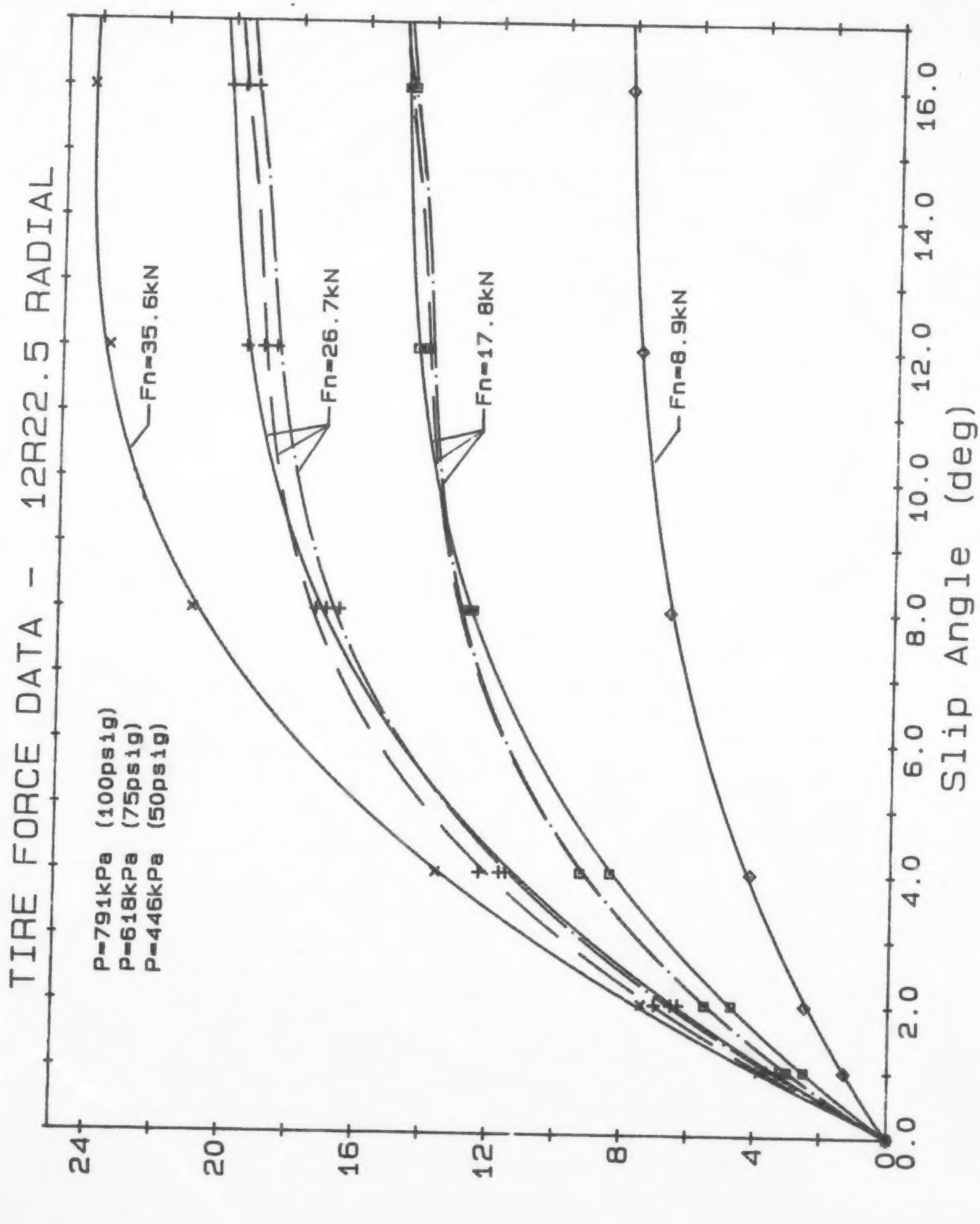


FIGURE 16: TIRE FORCE DATA FOR 12R22.5 RADIAL TIRE

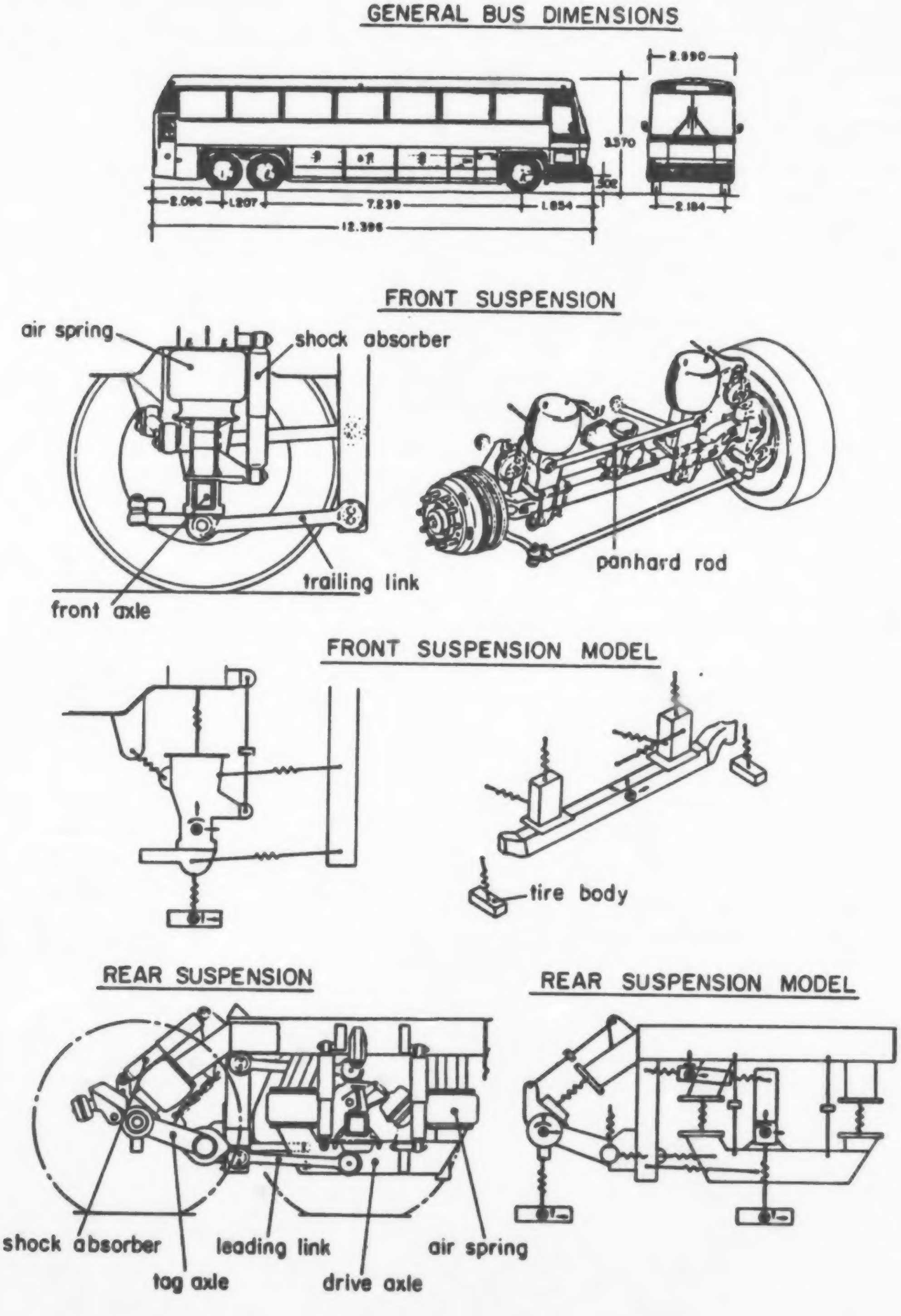

GENERAL BUS DIMENSIONS
2.990
3.370
302
2.096
1.207
7.239
1.854
12.396
2.184
FRONT SUSPENSION
air spring
shock absorber
trailing link
front axle
panhard rod
FRONT SUSPENSION MODEL
tire body
REAR SUSPENSION
shock absorber
tag axle
leading link
drive axle
air spring
REAR SUSPENSION MODEL

FIGURE 17: GENERAL LAYOUT OF BUS

the suspension arrangement. The air springs on the tag axles are generally maintained at constant pressure (35 psig) regardless of load, and may be easily depressurized if desired, allowing the axles to be chained up (leaving the bus in a 2-axle condition). Levelling valves on each side of the drive axle maintain the 4 drive air springs at constant height with changes in static load, as does a single levelling valve for the two front axle air springs. A total of 8 tires (2 front, 4 drive and 2 tag tires) support the vehicle on the roadway.

The computer model to describe this configuration is comprised of 13 rigid bodies as given in Table 3:

- o a single bus body, representing the body structure, engine, auxiliary systems, seats etc.
- o a front beam axle, including brake drums, spindles, airbag supports, etc.
- o a drive beam axle, including the differential, box frames forming an H-frame support for the air springs, brake drums, etc.
- o 2 tag axles, including pivot arms.
- o 2 diagonal upper links, between the body and drive axle.
- o 6 tire bodies, as described in Section 5.2.2, necessary in the simulation for road inputs.

The reference coordinate system for the model geometry is right-handed, the "x" longitudinal direction pointing forward, the "y" lateral direction pointing to the right, and the "z" vertical direction pointing downwards. The origin (o,o,o) is located at ground level at the intersection of the bus body longitudinal and lateral centerlines.

The carbody, each axle and the diagonal upper links are all given 6 degrees of freedom (DOF's), 3 translational (longitudinal, lateral, vertical) and 3 rotational (roll, pitch, yaw), while the tire bodies are given only the 3 translational degrees of freedom. This gives a total of 60 DOF's to the bus model.

The suspension elements connecting these rigid bodies are modelled as:

- o 65 linear spring-damper combinations, pointing in either the principal x, y or z directions, or torsionally about these directions.

RIGID BODY	C.G. LOCATION (m)			MASS (kg)	MOMENTS OF INERTIME ($kg-m^2$) ABOUT C.G.		
	x	y	z		Roll (IYY)	Pitch (IYY)	Yaw (IZZ)
Bus Body	-0.508	0.0203	-1.143	10650.	19300.	150100.	142500.
Front Axle	4.343	0.0	-0.502	916.	923.	8.0	921.
Drive Axle	-2.896	0.0	0.302	1806.	930.	52.3	945.
Tag Axle, Right	-4.102	0.933	-0.502	275.	9.5	4.0	11.5
Tag Axle, Left	-4.102	-0.933	-0.502	275.	9.5	4.0	11.5
Upper Diagonal Link, Drive Axle, Right	-3.237	0.355	-0.842	25.4	1.2	1.7	3.0
Upper Diagonal Link, Drive Axle, Left	-3.237	-0.355	-0.842	25.4	1.2	1.7	3.0
Front Tire Body, Right	4.343	1.092	0.0	10.	0.0	0.0	0.0
Front Tire Body, Left	4.343	-1.092	0.0	10.	0.0	0.0	0.0
Drive Tire Body (Dual), Right	-2.896	0.994	0.0	10.	0.0	0.0	0.0
Drive Tire Body (Dual), Left	-2.896	-0.994	0.0	10.	0.0	0.0	0.0
Tag Tire Body, Right	-4.125	1.071	0.0	10.	0.0	0.0	0.0
Tag Tire Body, Left	-4.125	-1.071	0.0	10.	0.0	0.0	0.0

TABLE 3 - MODEL RIGID BODIES, UNLADEN CONDITION

- 9 linear spring-damper combinations, pointing in a diagonal direction.

- 8 non-linear dampers (shock absorbers), pointing in a diagonal direction.

A comprehensive list of these 82 interconnection elements, their connection point locations and their force-deflection-velocity characteristics is included in Appendix A-2. A general breakdown of the interconnections is given in Table 4.

This data is arranged in an input file (listed in Appendix A-2) and processed by computer module A'GEM in preparation for the ride and handling simulations.

Any configuration or parametric change to the computer model is easily made in this data file. As examples, if the bus is loaded with passengers, adjustments to this input file involve:

- increasing the bus body mass and inertia to the appropriate values.

- increasing the air spring stiffnesses accordingly (on the front and drive axles only) since levelling valves on the bus maintain a constant height with load.

If low profile tires (eg. Michelin 275/80 R22.5) are substituted:

- each tire stiffness is adjusted accordingly.
- mass and inertia of each axle are lowered.
- the heights of all rigid body c.g.'s and suspension element connection points are lowered by 32 mm.

In this manner, the bus suspension system is examined at the component level. The bus model may be made as detailed as required to assess and make design changes.

DESCRIPTION	DIRECTIONS							TOTAL NUMBER
	LONGITUD X	LATERAL Y	VERTICAL Z	ROLL XX	PITCH YY	YAW ZZ	DIAGONAL	
8 tires	x	x	x					24
2 front, 4 drive air springs			x					6
2 tag air springs							x	2
2 upper, 2 lower trailing links, 1 panhard rod all with bushings, front							x	5
2 lower leading links with bushings, drive							x	2
4 bushings, upper links, drive axle	x	x	x	x	x	x		24
2 tag axle pivot pins	x	x	x	x		x		10
1 roll bar				x				1
Total Linear Elements								74
8 Shock absorbers, nonlinear force-velocity characteristics							x	8
Total Connection Elements								82 ===

TABLE 4 MODEL INTERCONNECTION ELEMENTS

5.4 REVIEW OF RIDE SIMULATION RESULTS

Data was gathered on the response of the actual Test Bus to several types of vertical input at the tires (Section 4.5). These vertical inputs (or representations) were entered into the ride simulation modules, and the response of the Test Bus computer model to these inputs were determined and compared with the test results.

5.4.1 Defined Vertical Input, Shaker Tests

The response of the bus to the very well-defined vertical inputs on the NRC shaker were most valuable in examining the computer model and assessing the suspension component vertical characteristics. Several iterations on the component properties were made in developing the computer model.

In Figure 18, a comparison is shown of the bus body accelerations in time at 3 locations within the body as the drive and tag tires are simultaneously subjected to a sinusoidal displacement of ±12.7 mm. in phase at frequency of 1.8 Hz. The bus in this instance was fully loaded (13,750 kg) and in its normal configuration. The identical sine wave input to these tire bodies was used in the simulation to derive the model results.

Axle accelerations for the same input are compared in Figure 19. The apparent higher frequency variation superimposed on these accelerations were noticeable in the test results. Using the computer model, these variations were found to arise from the non-linear force characteristics of the shock absorbers between compression and extension.

In Figure 20, the vertical displacements measured between the bus body and the front and rear axles are compared with those predicted by the computer model.

Comparisons between the test results and the simulation were made for this in-phase input to the drive and tag wheels over a frequency range from 0.5 Hz to 15.0 Hz. The normalized response of the bus body at the c.g. to this input is shown as a function of frequency in Figure 21. The peak response at this location was measured and is predicted in the simulation to be about 1.8 Hz as shown. It was of particular interest that, using the eigenvalue analysis option in BUSRID (uses only the linear equations of the bus model), the characteristic or natural frequencies for bus body pitch (1.2 Hz) and body bounce (1.5 Hz) in this configuration were somewhat lower than the peak response value measured. With the time domain simulation, non-linear forces through the shock absorbers (particularly friction present in both compression and extension) are seen to result in this increase in predicted peak response frequency.

To examine the roll properties for the bus and model, the drive and tag tires were subjected to sinusoidal vertical displacements, 180° out of phase, side to side.

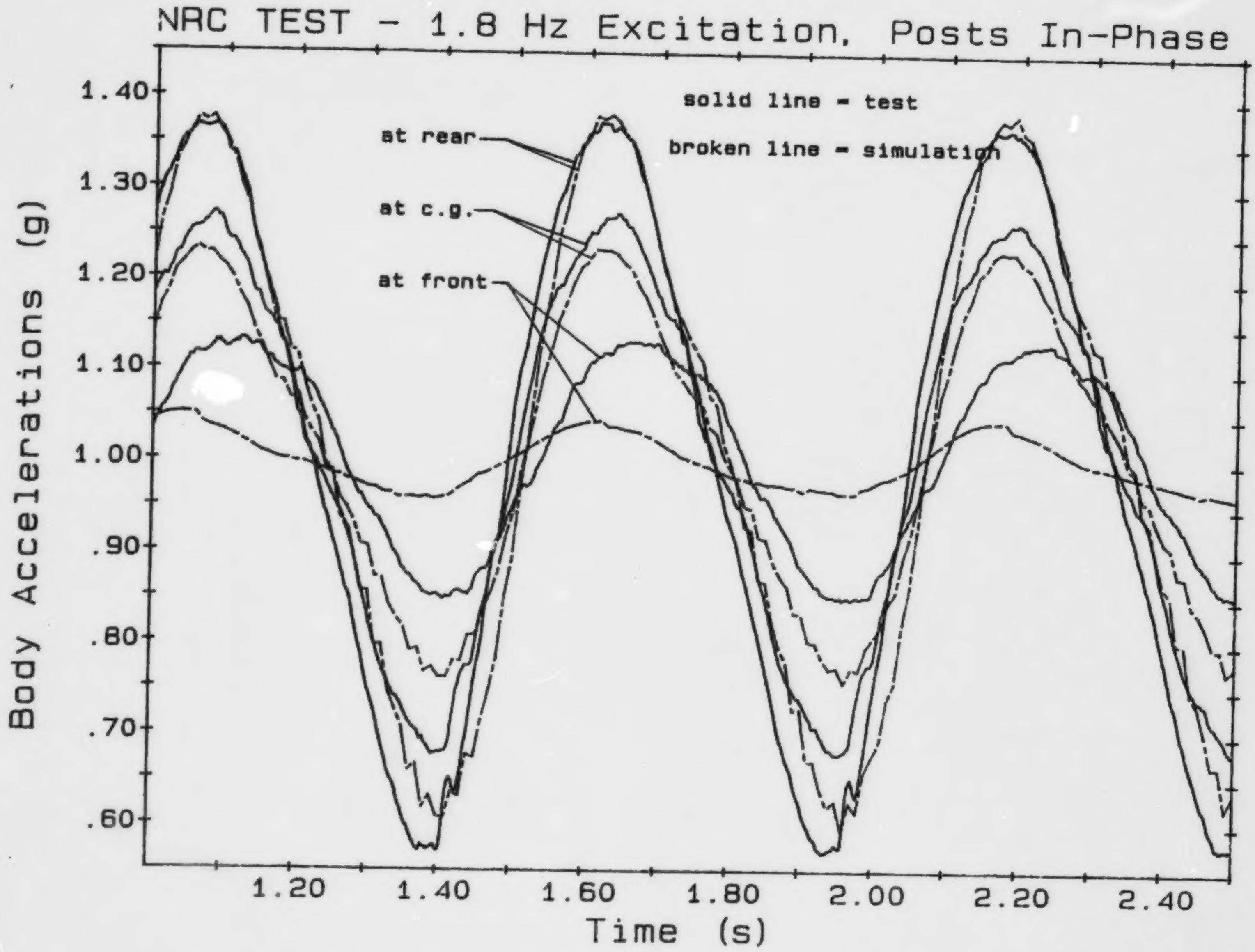


FIGURE 18: COMPARISON OF BUS BODY ACCELERATIONS FOR TEST AND SIMULATION

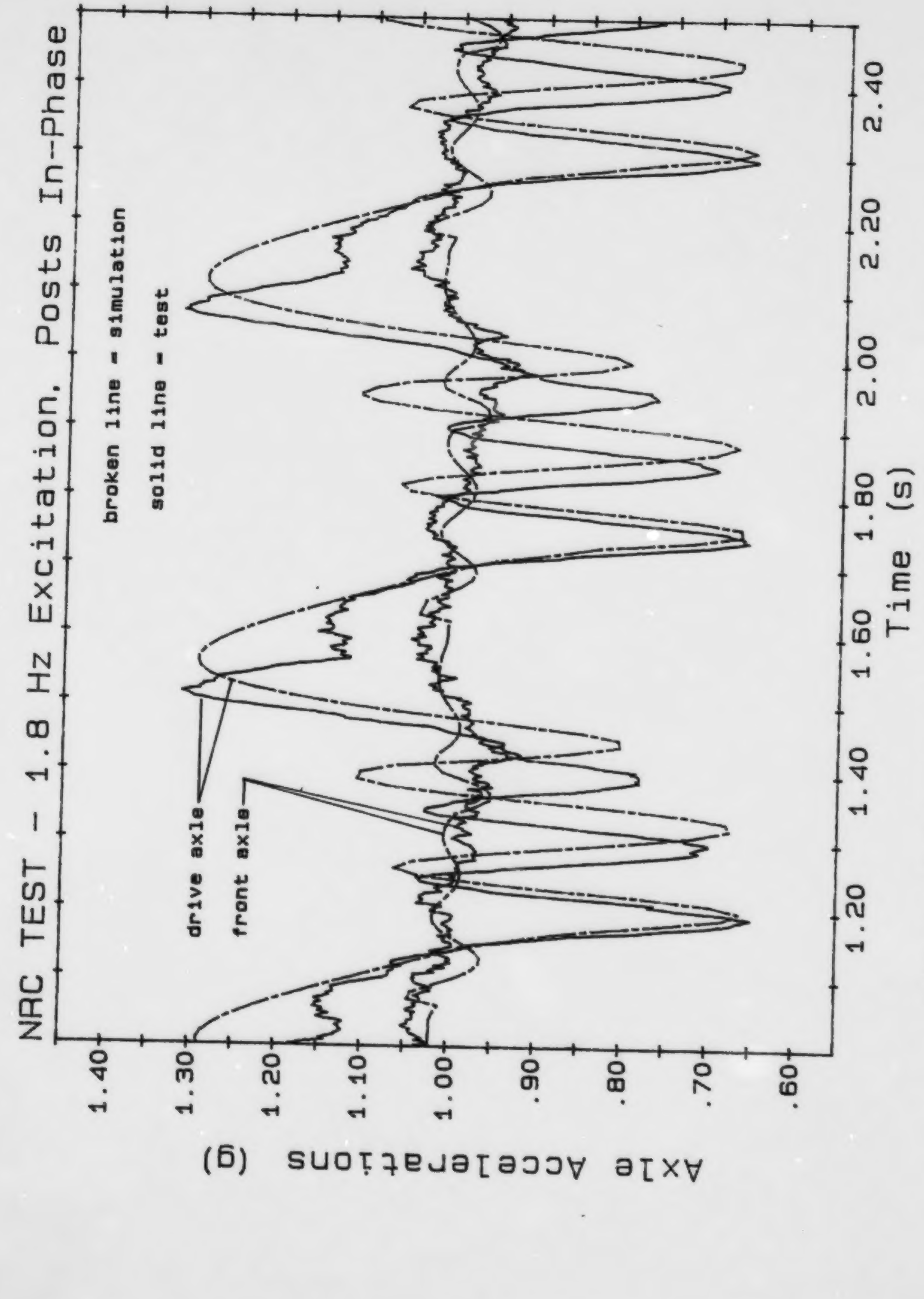


FIGURE 19: COMPARISON OF AXLE ACCELERATIONS FOR TEST AND SIMULATION

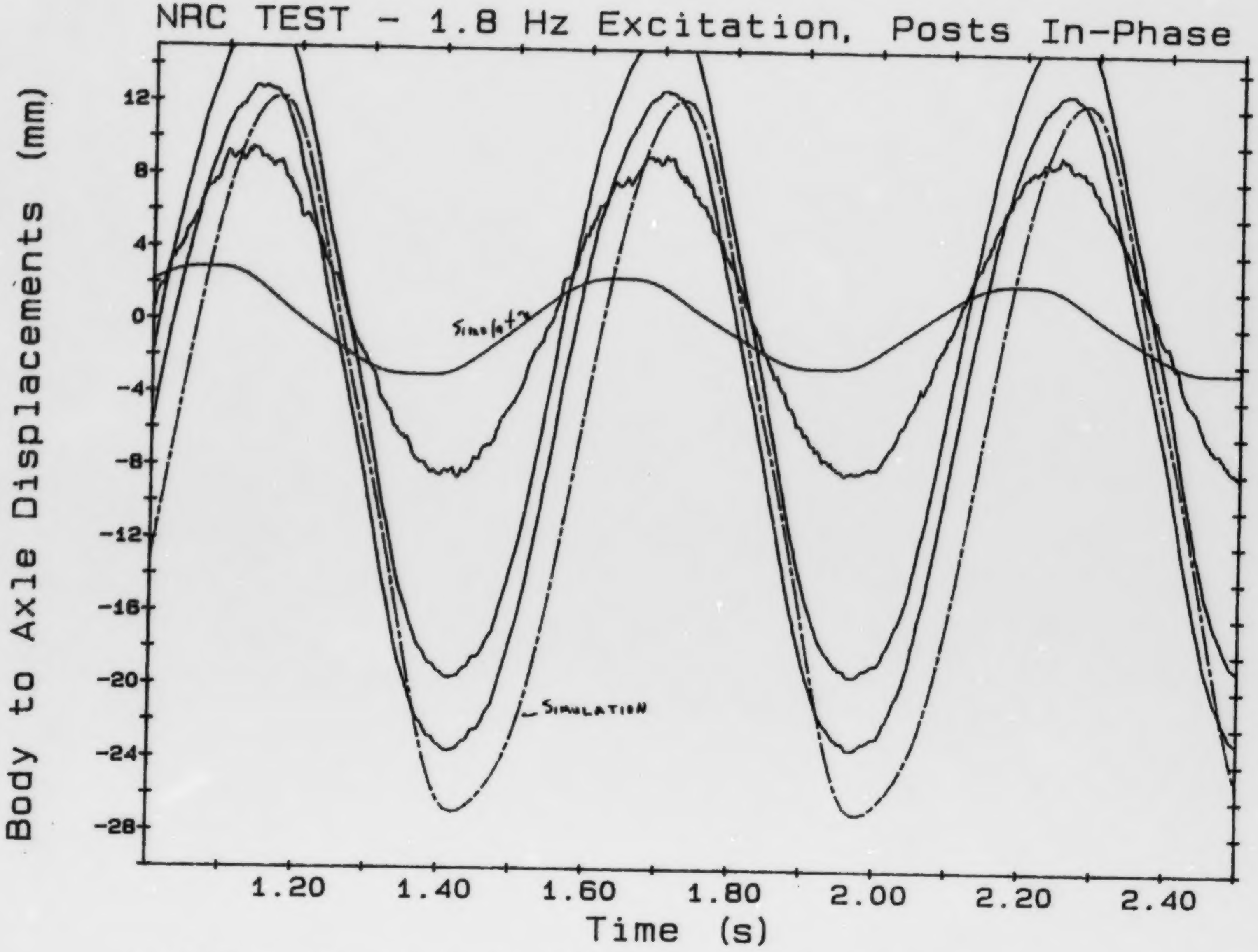


FIGURE 20: COMPARISON OF AXLE TO BODY DISPLACEMENTS FOR TEST AND SIMULATION

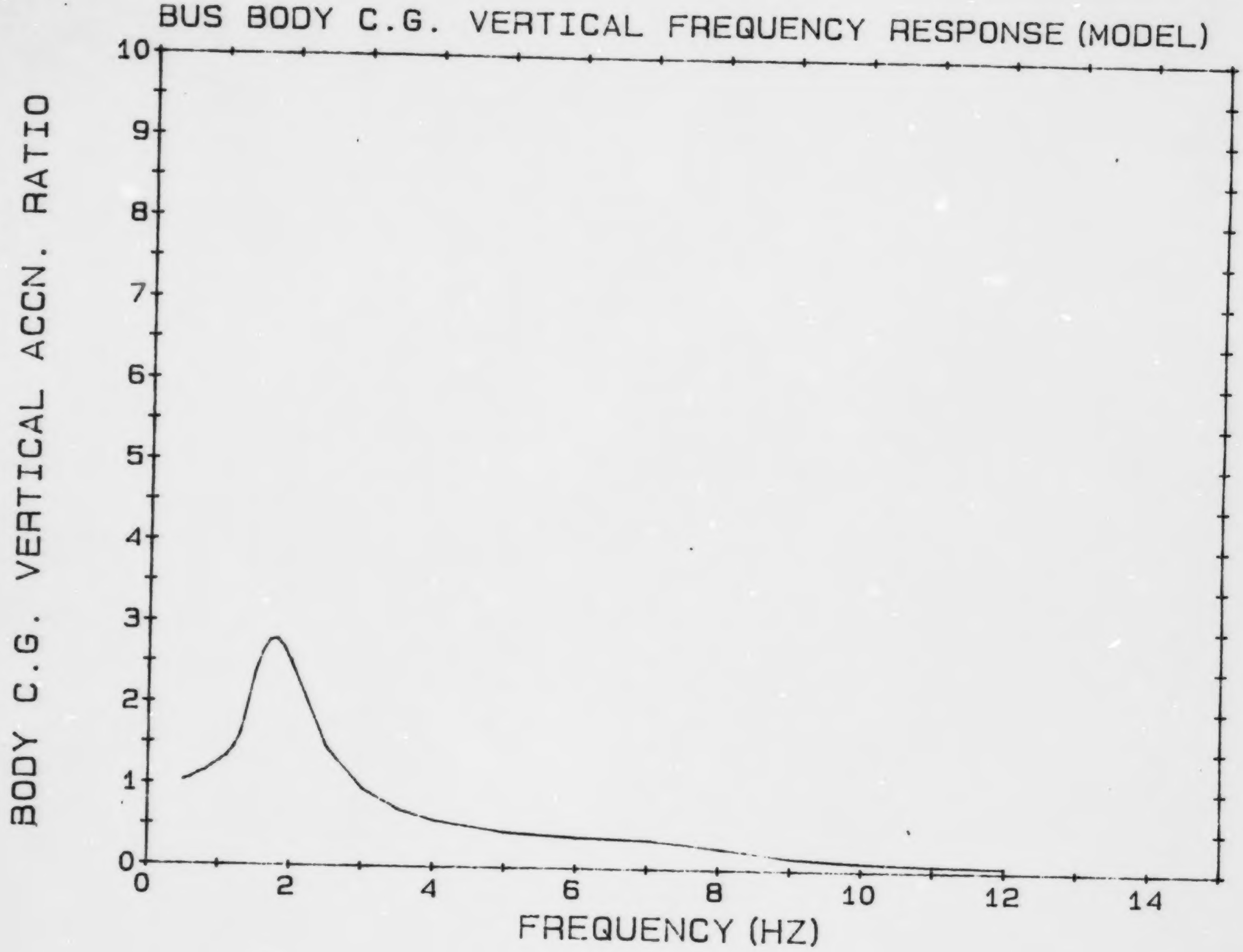


FIGURE 21: BODY C OF 6 VERTICAL ACCELERATION AS A FUNCTION OF FREQUENCY

In Figures 22 and 23, the bus body accelerations and vertical displacements between body and rear axle are shown and compared with simulation predictions for input vertical displacements of 12.7 mm (.5"), out of phase. In this case the tag axle air springs were deflated in both the tests and simulation. As in the case of the in-phase input shown in Figure 20 the agreement is less good at the front axle.

5.4.2 Vertical Step Input

The vertical acceleration response of the Test Bus, driven over a vertical "bump" at 22.4 m/s (50 mph), is shown in Figure 24, for a case where the bus was unloaded and with standard tires. The simulated time response is also shown.

5.5 REVIEW OF HANDLING SIMULATION RESULTS

The handling behaviour of the Test Bus when subjected to various steering inputs was examined using the bus model and compared with the test results. In each case, the front steered wheel angles as measured in a test (as a function of time) were input to the simulation together with the measured forward speed. The response of the bus as modelled was then compared with the response as measured in the test.

5.5.1 Steady State Results

Handling tests were conducted in which the bus was driven at several constant speeds on a path of constant radius. At each speed, steady state conditions (constant in time) were reached and measurements made of the steering hand-wheel angle and the front steered wheel angles required to follow this circular path. The bus body lateral acceleration, sideslip angle, roll angle and front axle angle resulting from each manoeuvre were monitored. These steady-state measurements enabled a further tuning of the suspension component lateral characteristics in the bus model, including the tire force curves.

In Figure 25, the measured front wheel angles and the steering hand-wheel angle are shown as a function of bus forward speed travelling in both the clockwise and counterclockwise directions on a 40 m radius circle. In this case, the bus was installed with low profile tires and was fully loaded. Since computer module BUSHAN is a time domain simulation, it was necessary to input the measured front wheel angle and forward speed for each case separately, and run the simulation until steady state conditions were achieved.

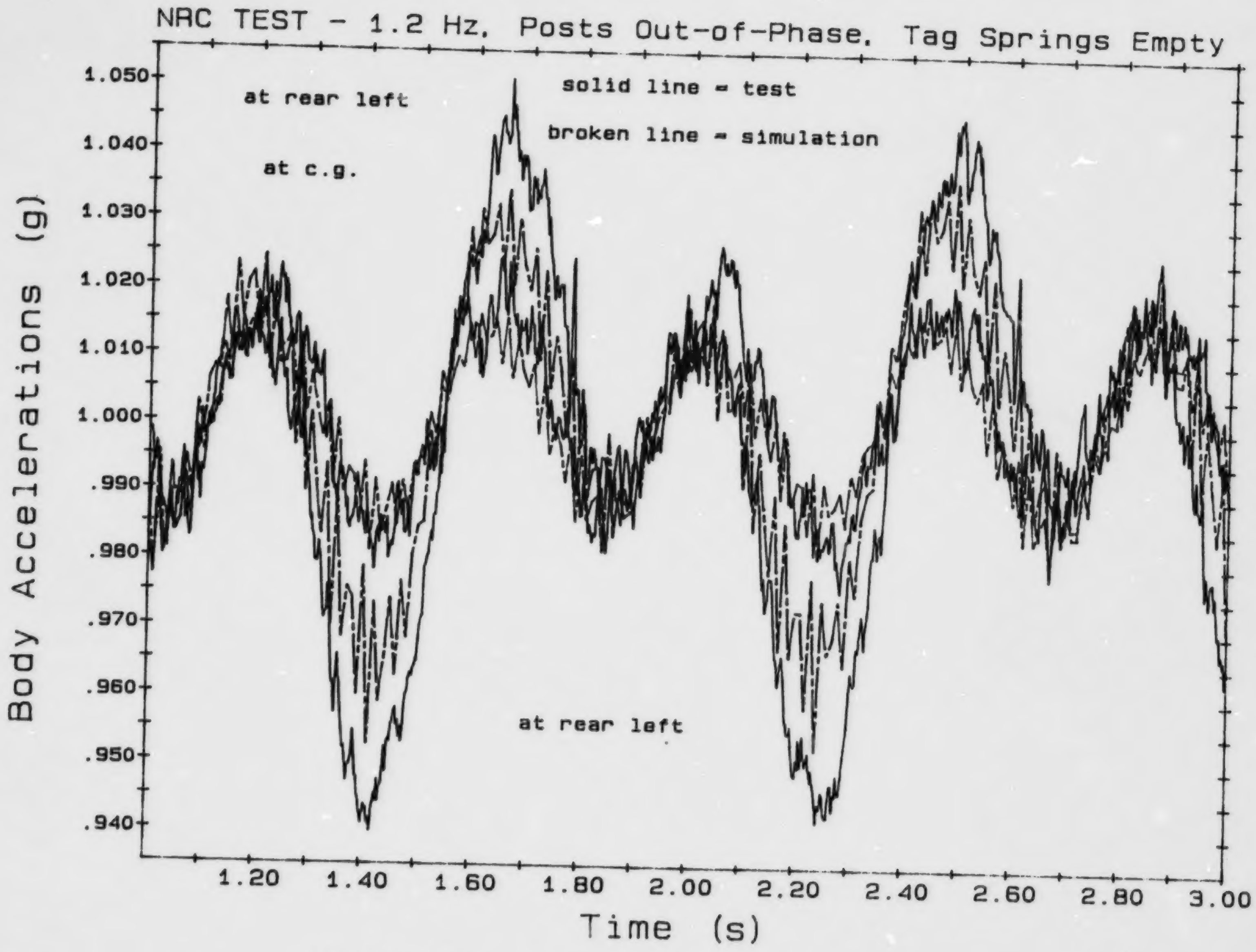


FIGURE 22: BUS BODY ACCELERATIONS FOR OUT-OF-PHASE INPUT

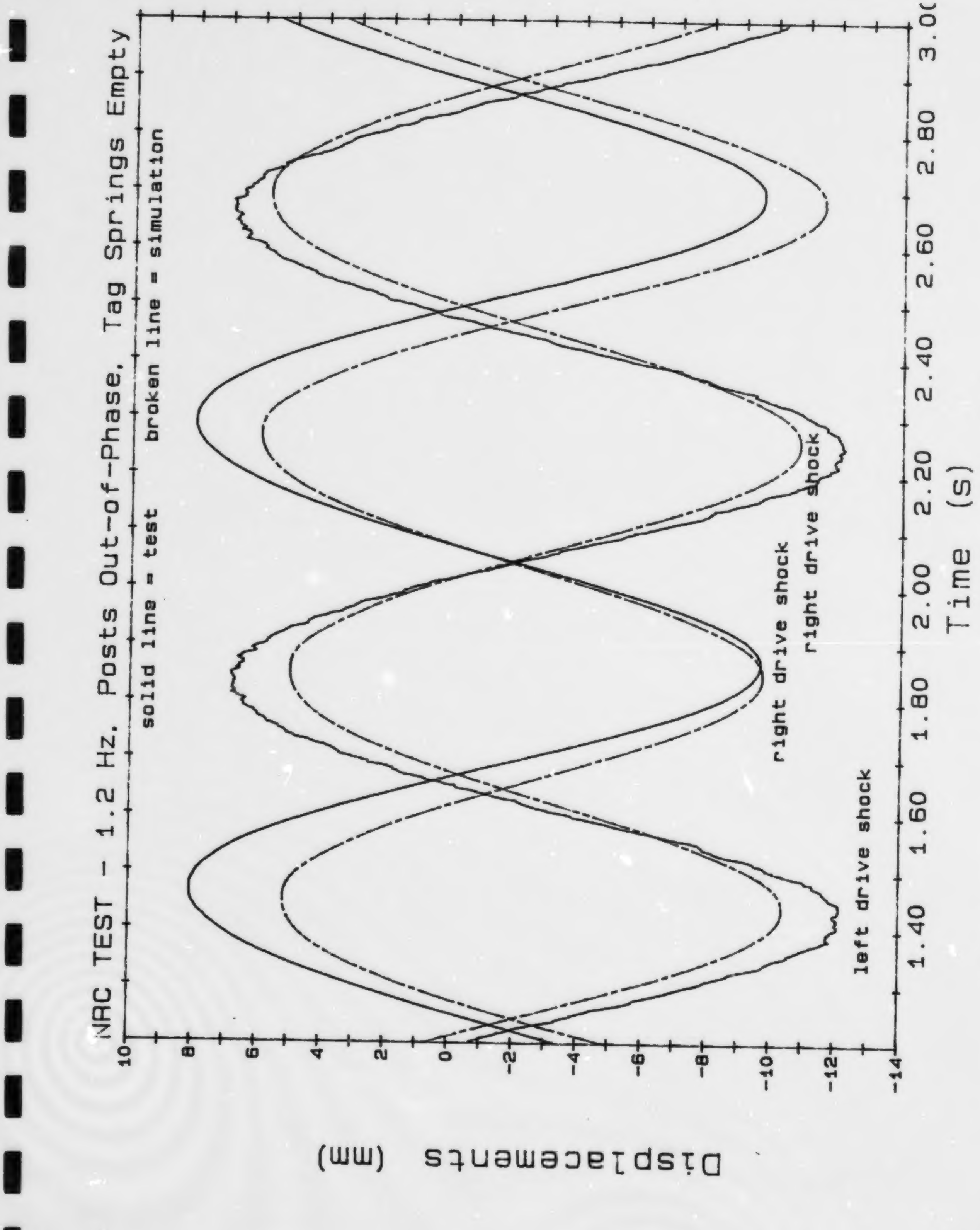


FIGURE 23: REAR AXLE TO BODY VERTICAL DISPLACEMENT FOR OUT-OF-PHASE INPUT

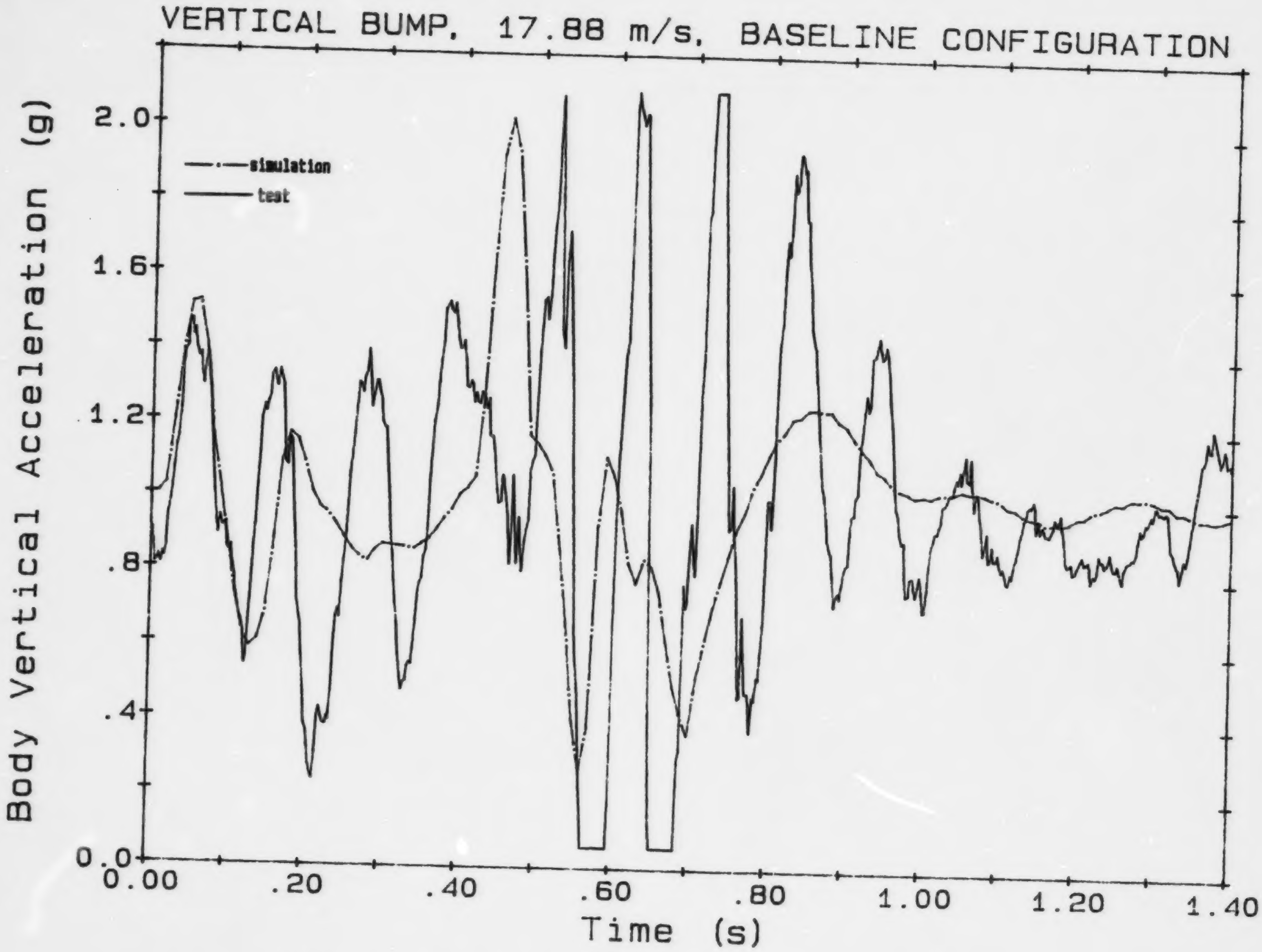


FIGURE 24: VERTICAL ACCELERATION RESPONSE OF BUS BODY TO STEP INPUT

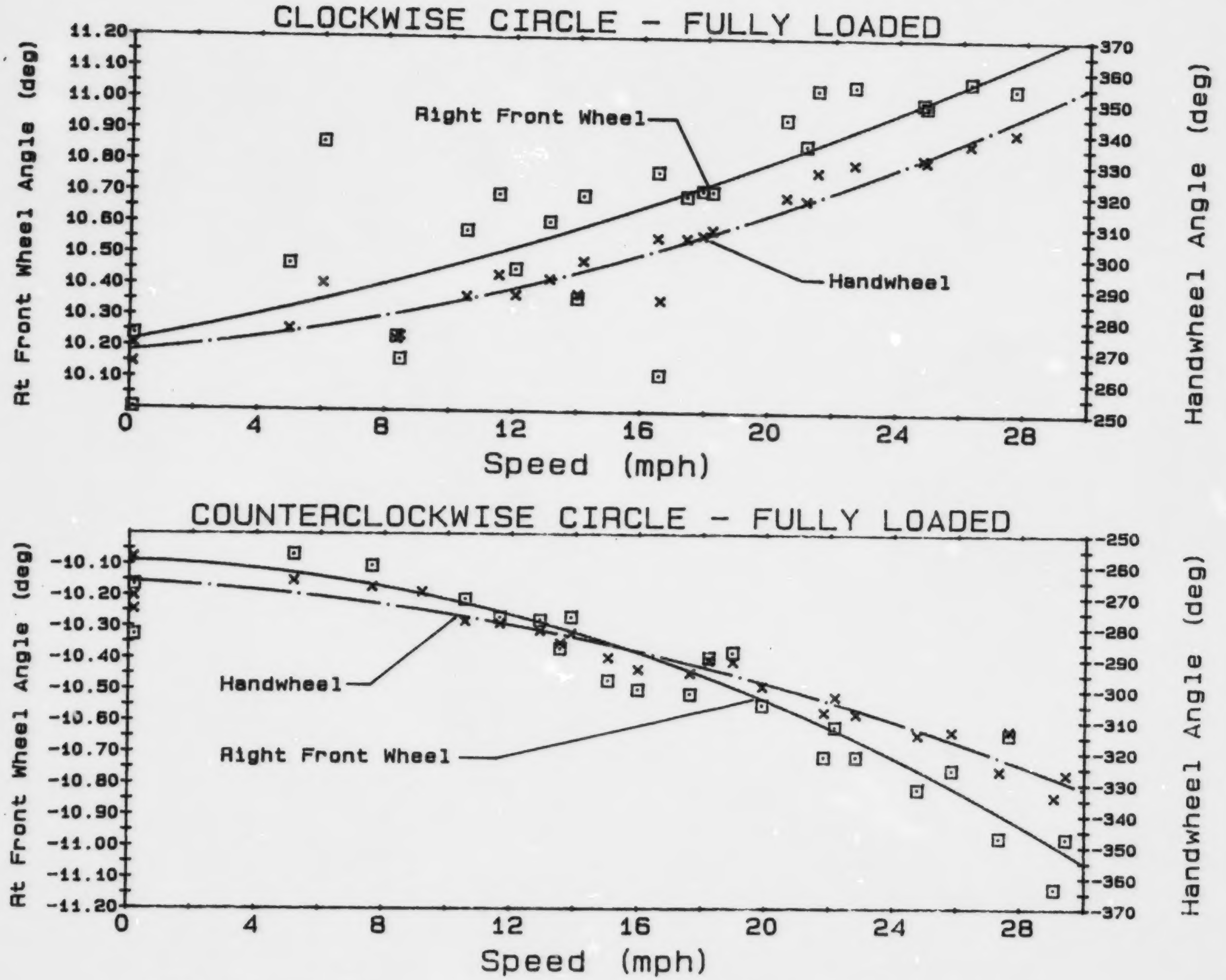


FIGURE 25: FRONT WHEEL AND HANDWHEEL ANGLES AS A FUNCTION OF SPEED: 40 M CIRCLE

Steady state handling results are generally presented (Ref. 3) as a function of the body lateral acceleration at the c.g. In Figure 26, the measured and simulated wheel angle characteristics are shown for the tests defined in Figure 25. The body roll and body sideslip characteristics are shown in Figures 27 and 28 respectively, comparing the simulation results with test.

5.5.2 Step Input Steer

The transient response of the Test Bus to a step steering input was examined by making right and left turns while travelling at a forward speed of about 14.75 m/s (33 mph). As exemplified in Figure 29, the front wheel angle was turned from zero to about +10° (to the right) in a time of about 0.9 seconds and held at this value. For each given test, the measured step input signal for wheel angle was entered into the handling simulation.

In Figure 29, the measured lateral acceleration of the bus body is also shown in response to the steering input and compared with the simulated response. The case shown is for the fully loaded bus with low profile tires. The response and peak response times, are representative of the rear-engined highway bus. In Figures 30 and 31, the body yaw rate and roll rate responses are also compared with the simulated results.

For a different bus configuration, a comparative case is shown in Figures 32, 33 and 34 in which the bus was unloaded and with standard bus tires. Again, the simulation predicts well the measured response for this configuration.

5.5.3 Response to Random Steer Input

Continuous time response of the Test Bus to random steer input was also measured by driving the vehicle at a forward speed of 22.8 m/s (50 mph) and making continuous inputs to the steering hand-wheel. Figure 35 shows a representative time history of the right front wheel angle measured during a test where the bus was fully loaded with the tag axles lifted. The front wheel angle signal was input to the handling simulation for this configuration, and the results are compared with values measured in the test for body lateral acceleration, yaw rate and roll rate in Figures 36, 37 and 38 respectively.

Several test runs were carried out for each of several bus configurations. Based on the assumption of linear vehicle behaviour, a frequency domain analysis of the time history results could be made for each bus configuration, showing, for example, lateral acceleration and yaw rate related to steering wheel angle. However, at the present time, only the time histories were compared between test and simulation in validating the computer model.

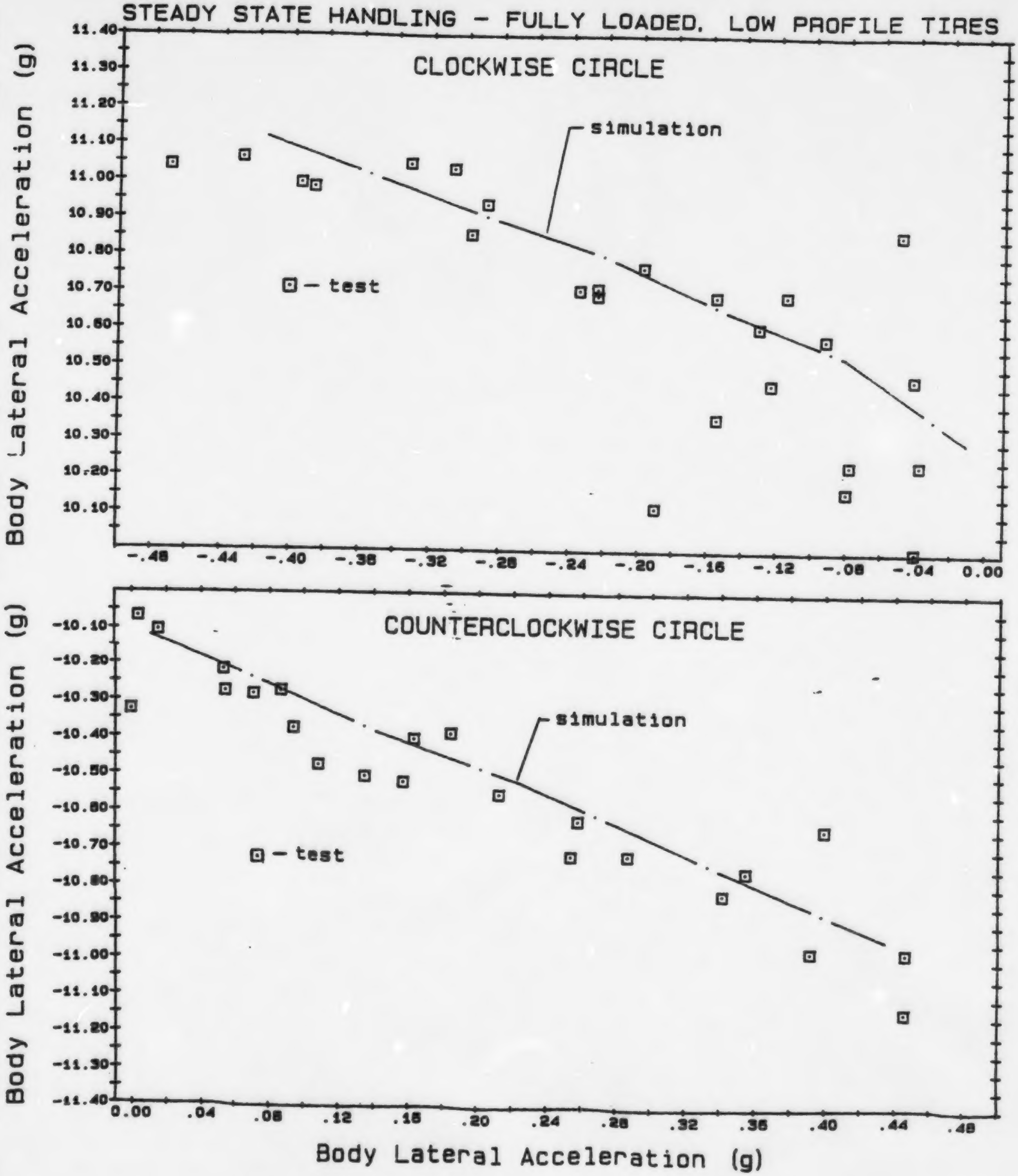


FIGURE 26: LATERAL ACCELERATION AS A FUNCTION OF FRONT WHEEL ANGLE FOR 40 M CIRCLE

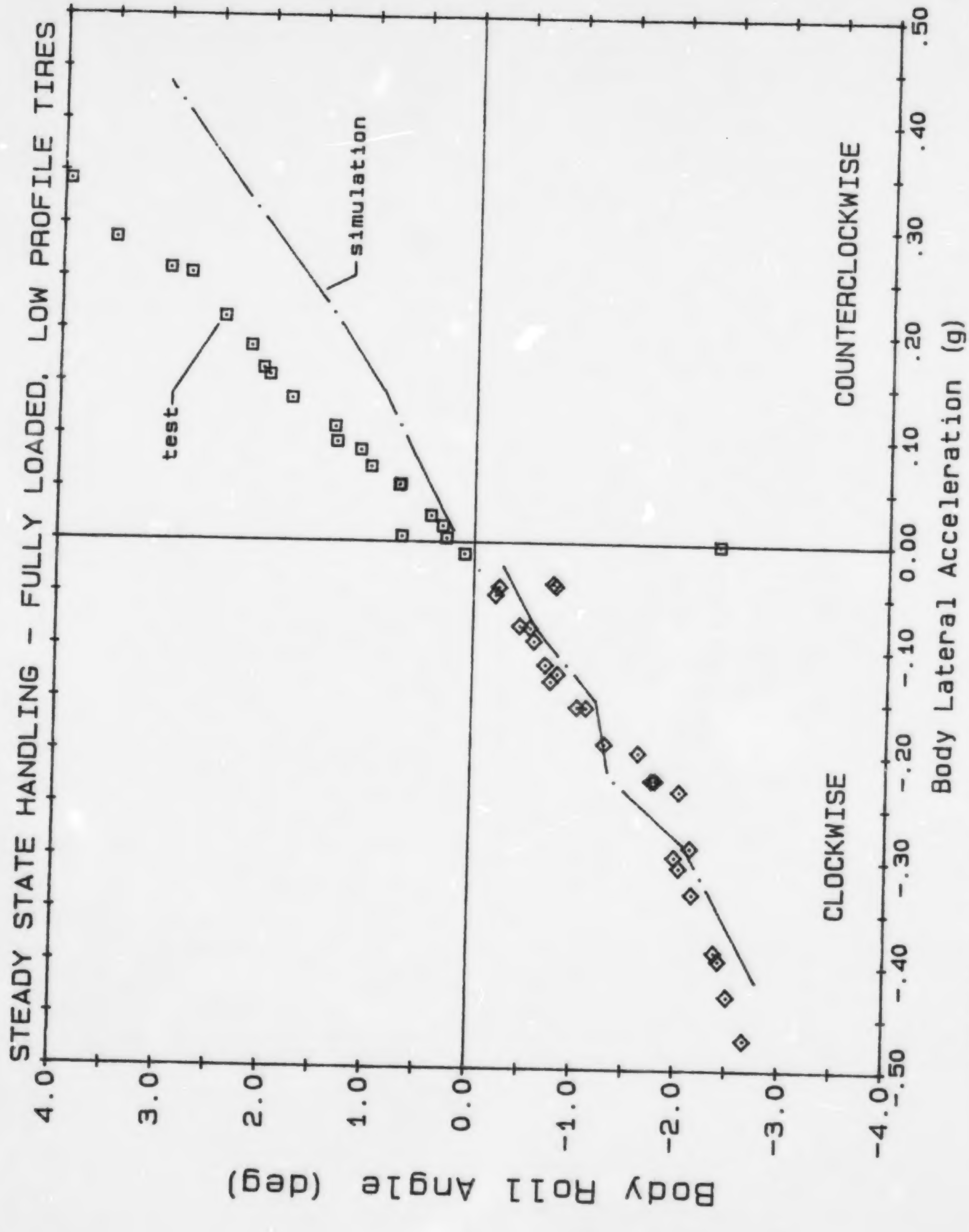


FIGURE 27: ROLL ANGLE AS A FUNCTION OF LATERAL ACCELERATION FOR A 40 M CIRCLE

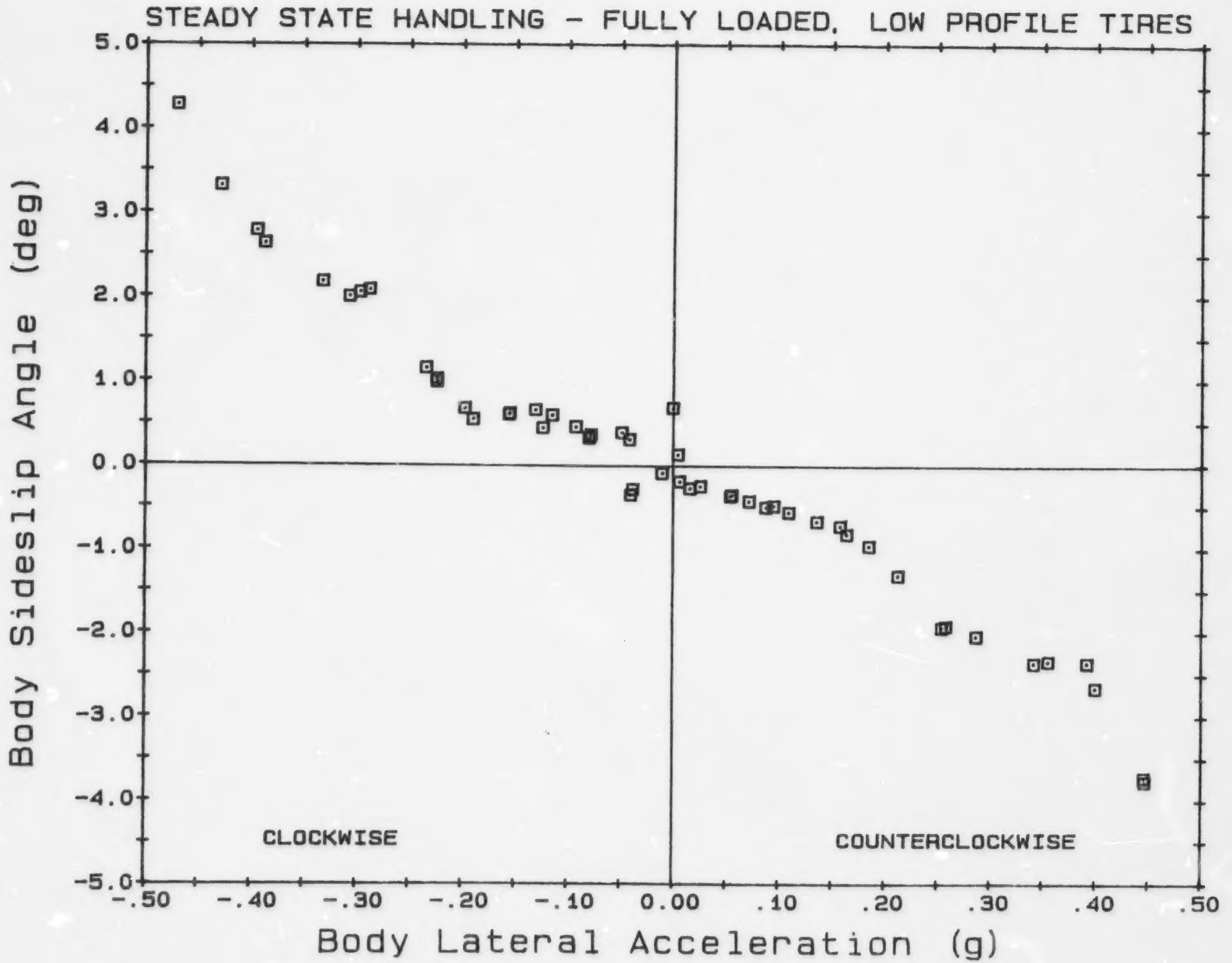
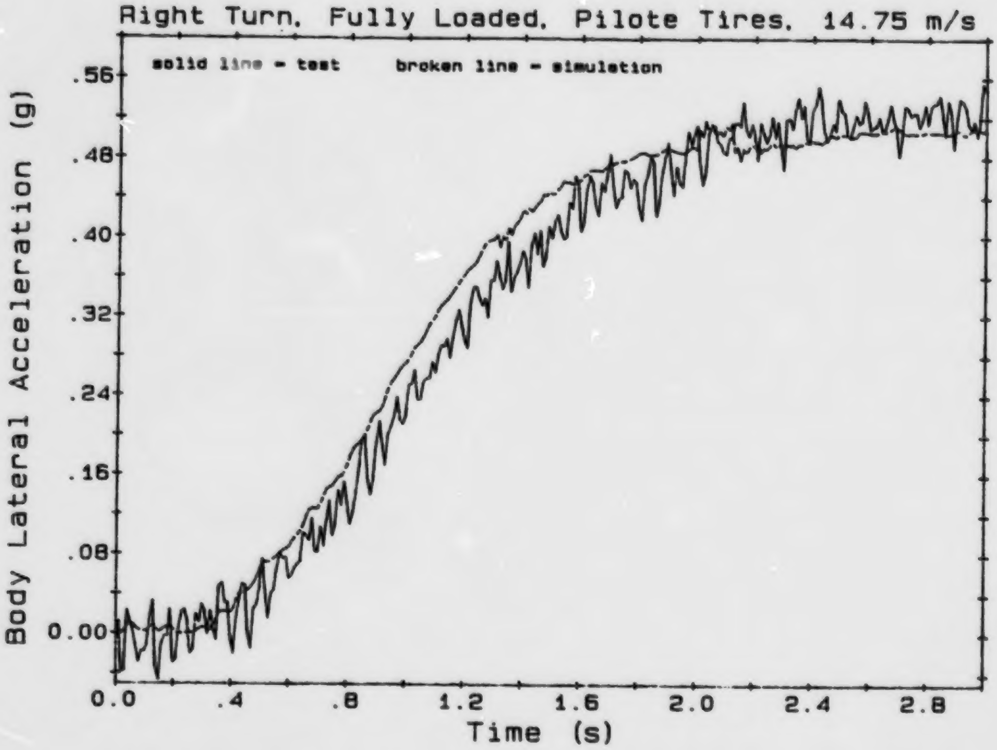


FIGURE 28: YAW ANGLE AS A FUNCTION OF LATERAL ACCELERATION FOR A 40 M CIRCLE



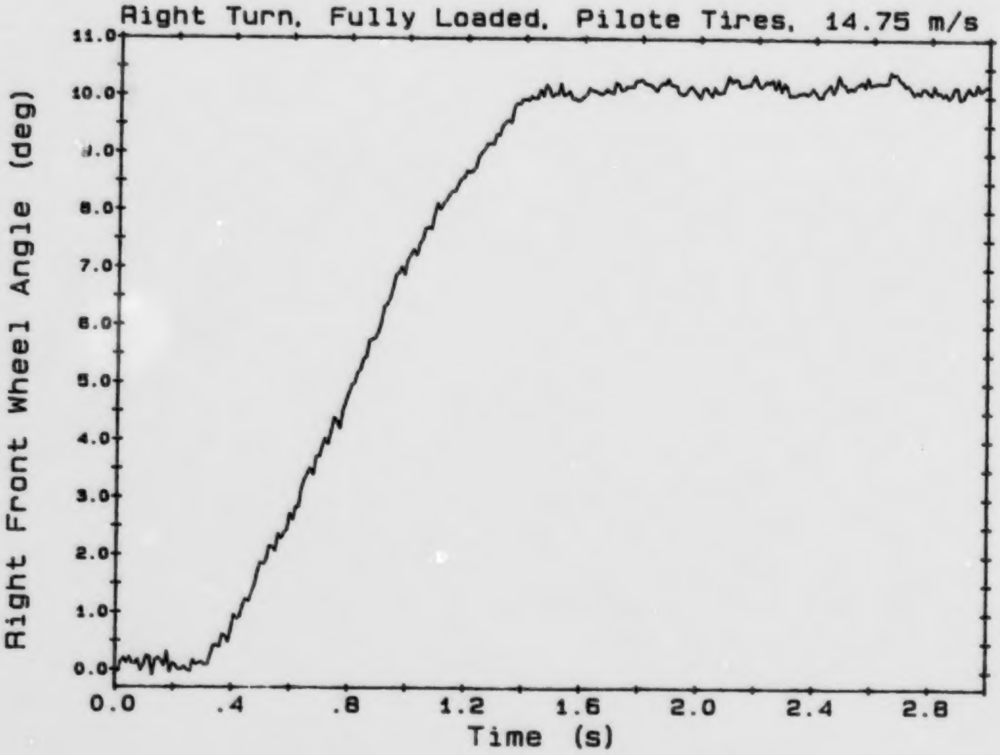


FIGURE 29: TRANSIENT LATERAL ACCELERATION RESPONSE TO STEP STEERING INPUT FULLY LOADED, PILOTE TIRES

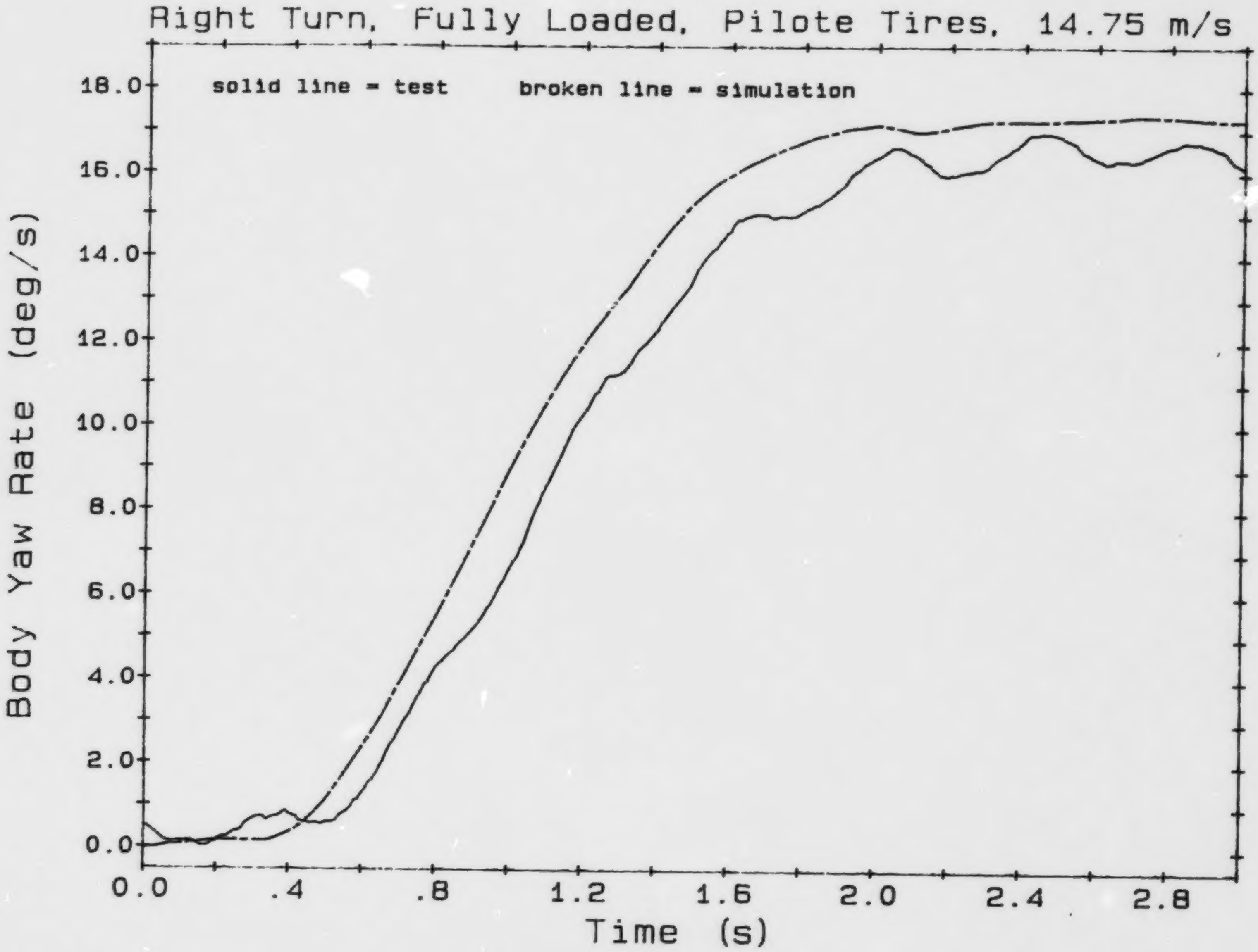


FIGURE 30: TRANSIENT BODY YAW RATE RESPONSE TO STEP STEERING INPUT - FULLY LOADED, PILOTE TIRES

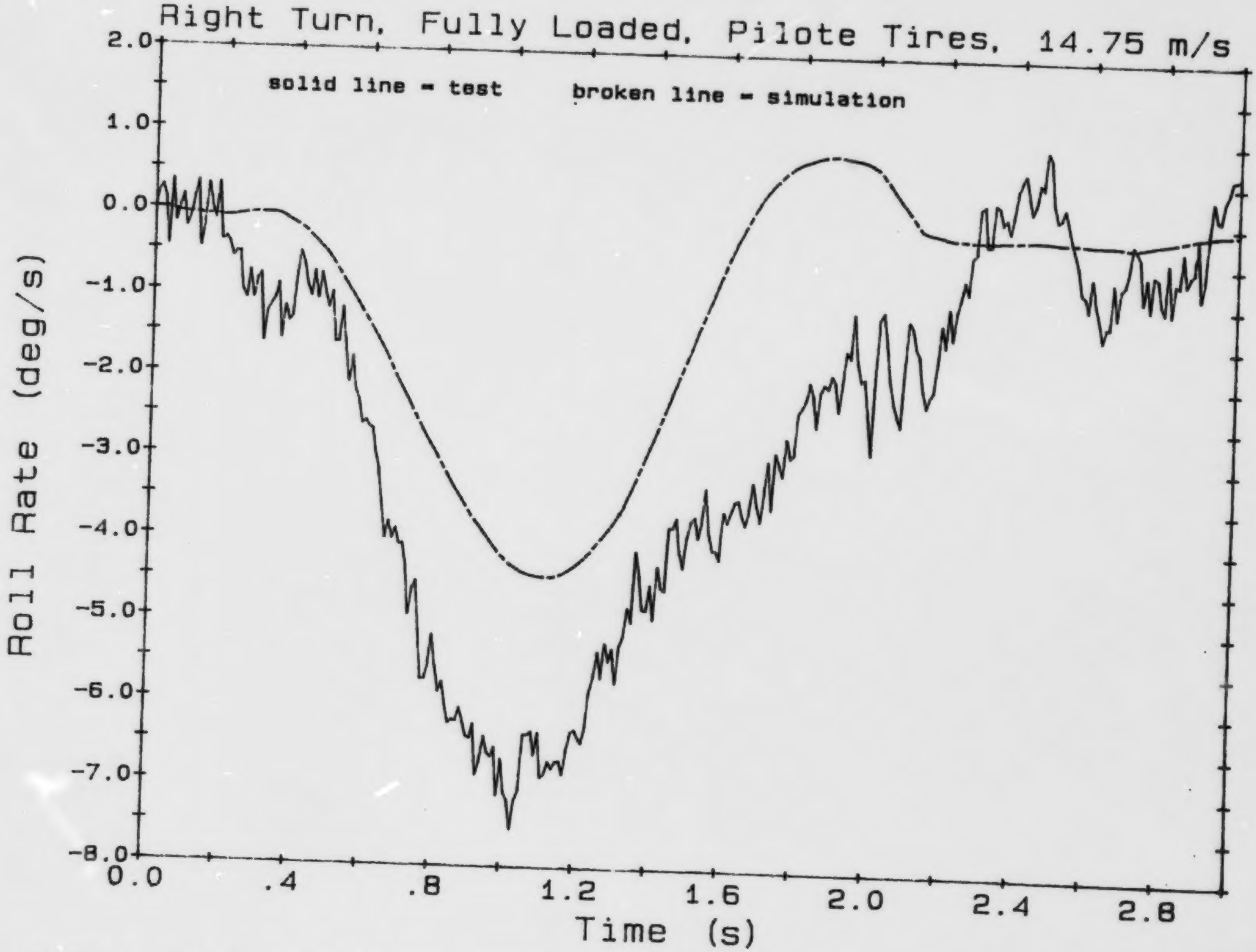


FIGURE 31: TRANSIENT BODY ROLL RATE RESPONSE TO STEP STEERING INPUT - FULLY LOADED, PILOTE TIRES

K8-10498

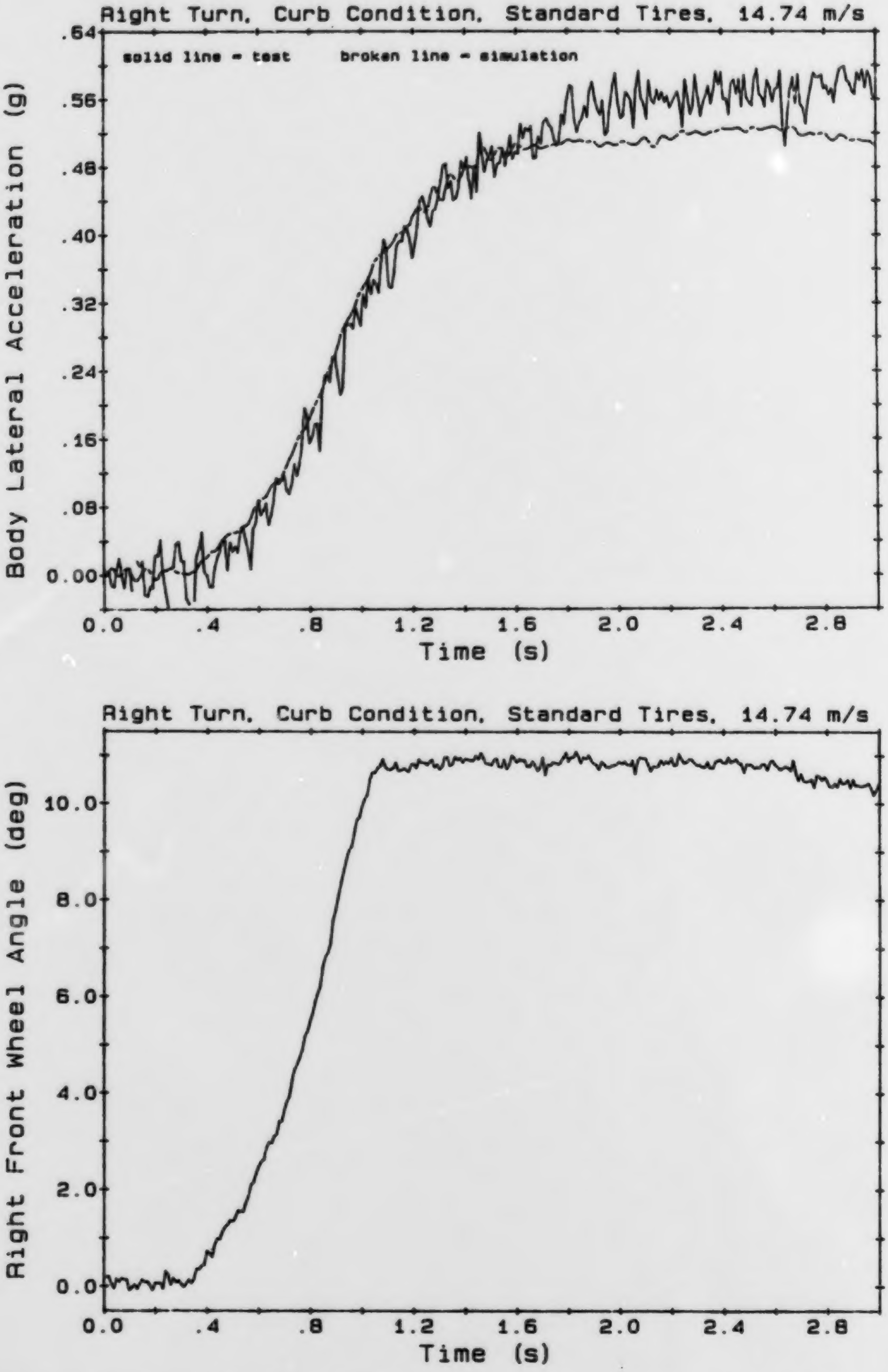


FIGURE 32: TRANSIENT LATERAL ACCELERATION RESPONSE TO STEP STEERING INPUT - CURB WEIGHT, STANDARD TIRES

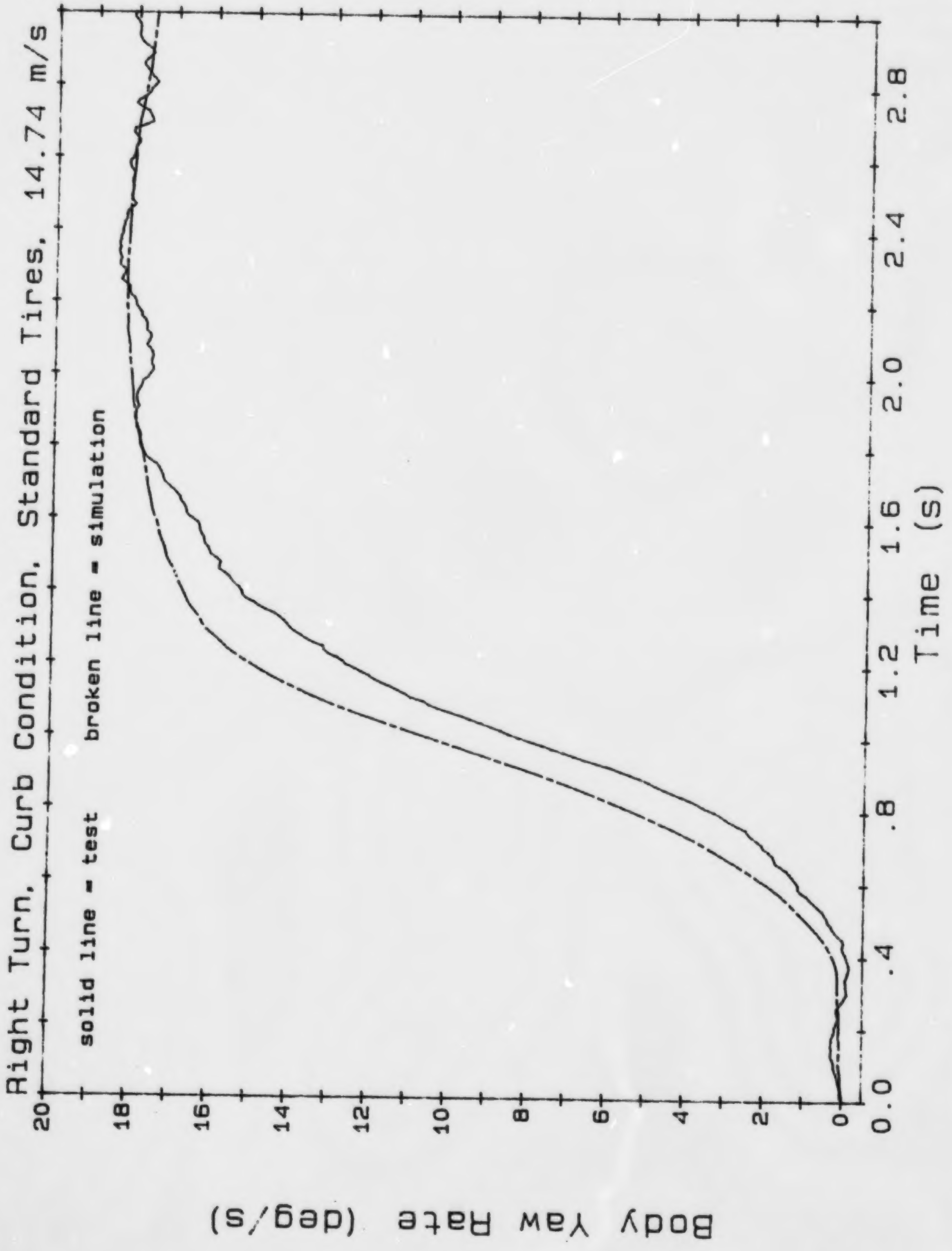


FIGURE 33: TRANSIENT BODY YAW RATE RESPONSE TO STEP STEERING INPUT - CURB WEIGHT, STANDARD TIRES

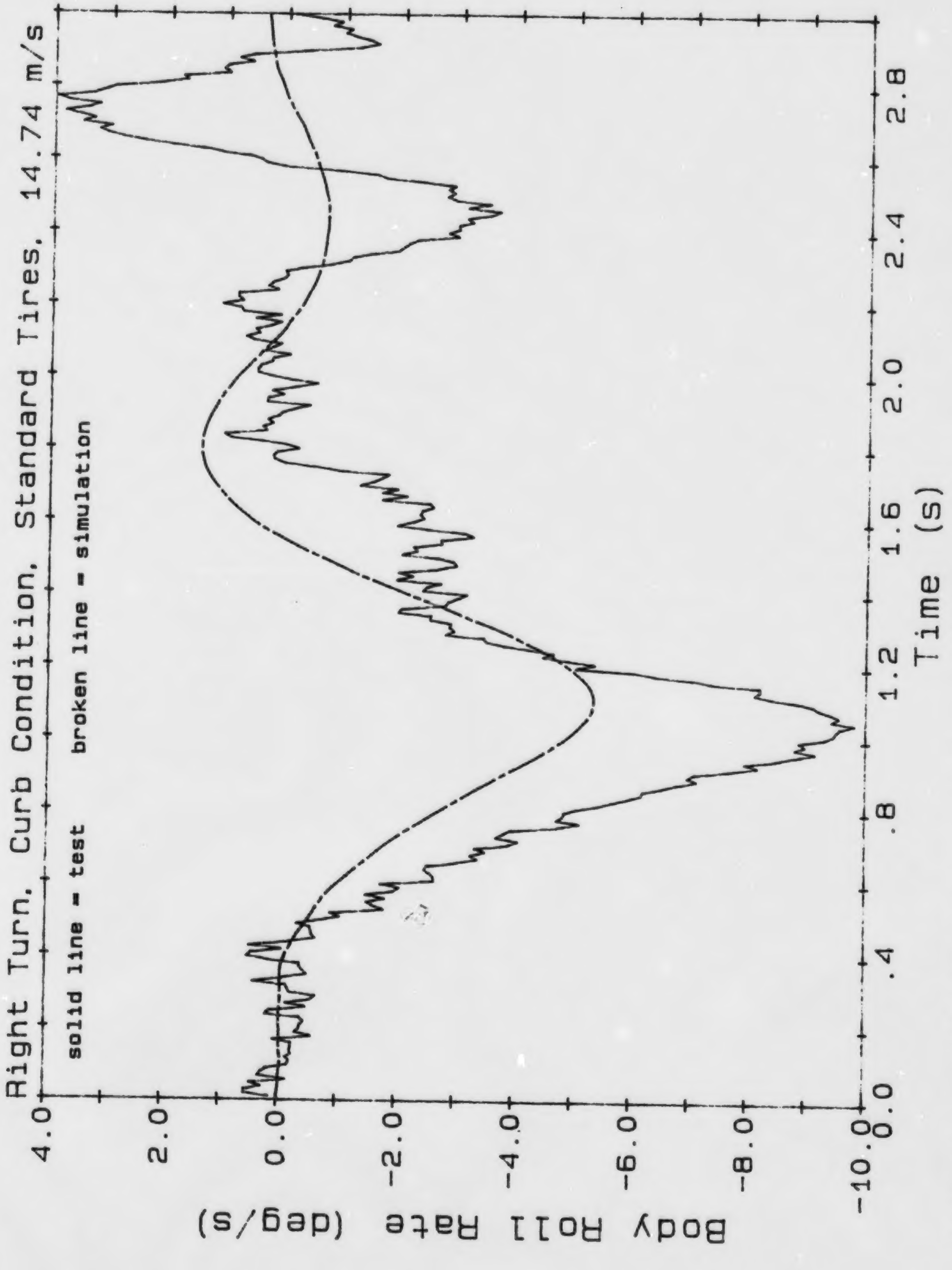


FIGURE 34: TRANSIENT BODY ROLL RATE RESPONSE TO STEP STEERING INPUT - CURB WEIGHT, STANDARD TIRES

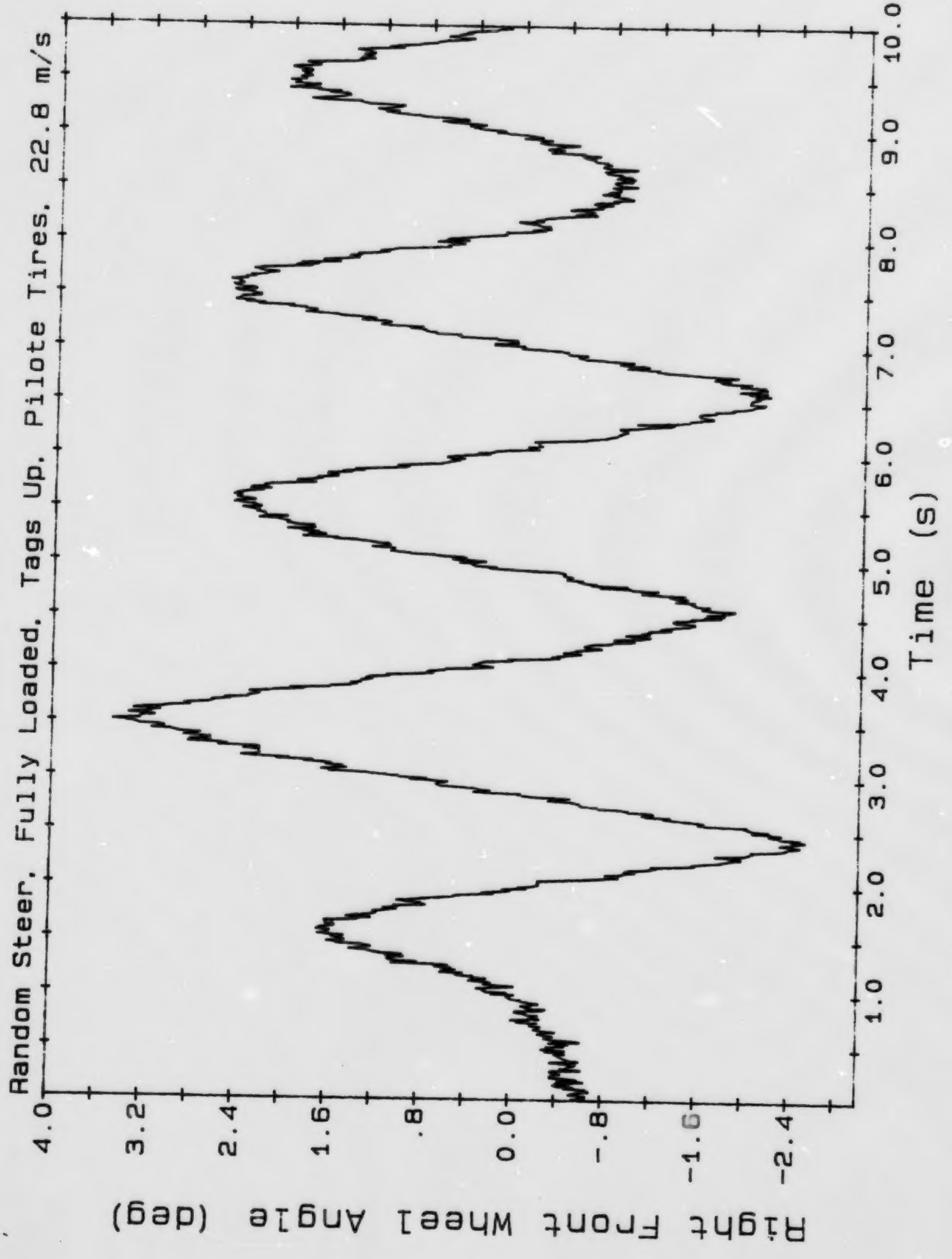


FIGURE 35: TYPICAL FRONT WHEEL ANGLE FOR RANDOM STEER TEST

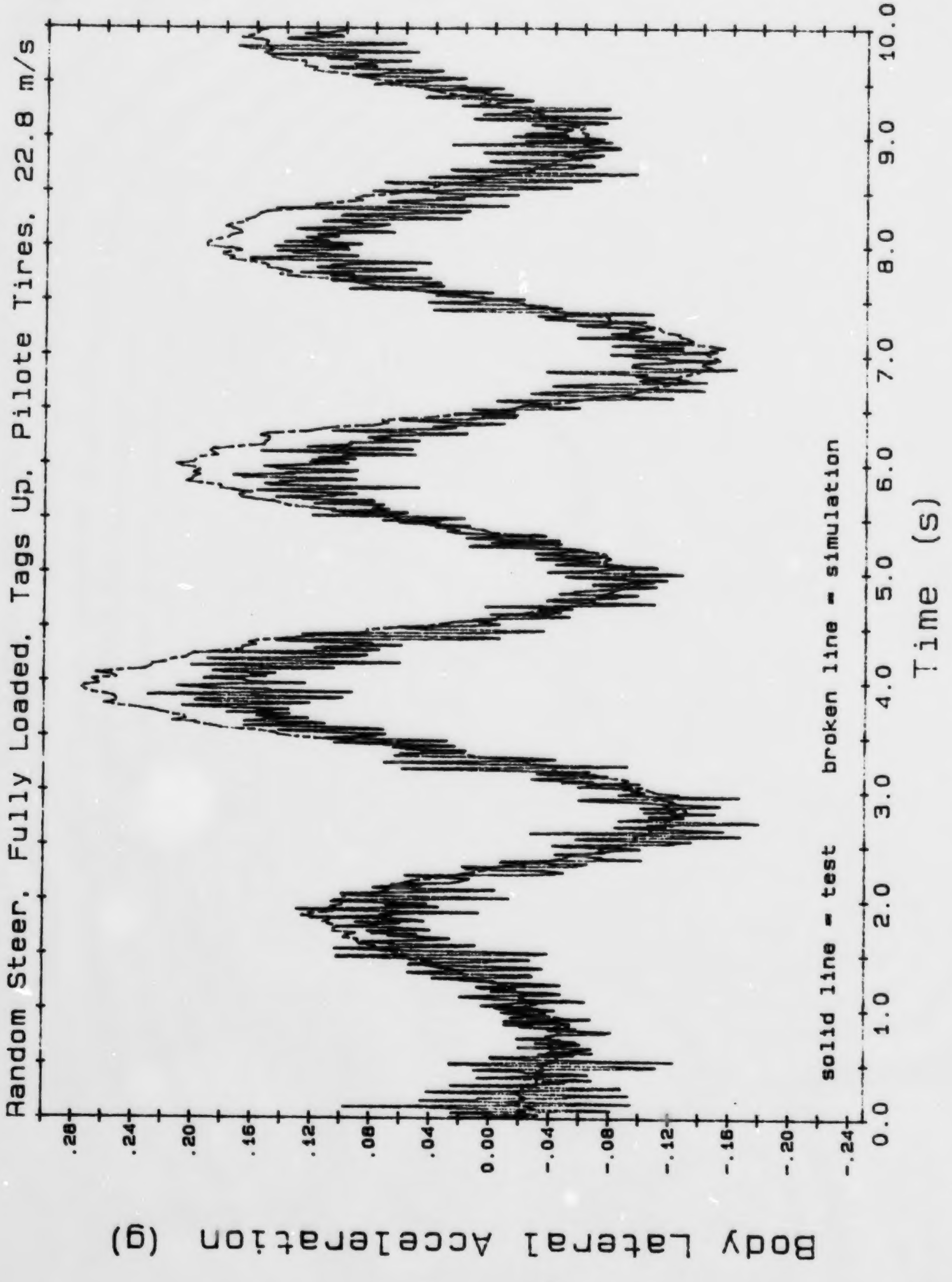


FIGURE 36: BODY LATERAL ACCELERATION RESPONSE TO RANDOM STEER INPUT

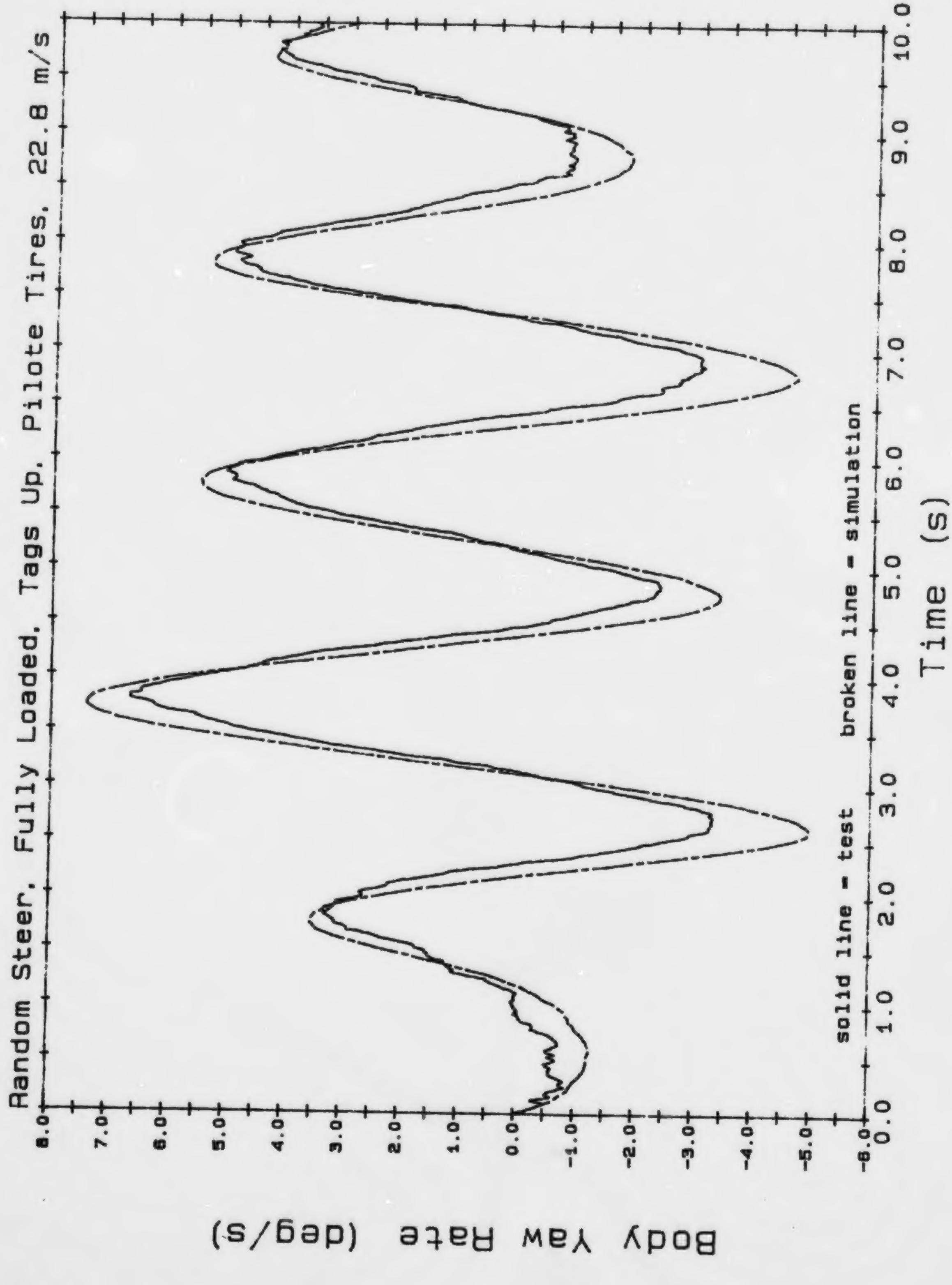


FIGURE 37: BODY YAW RATE RESPONSE TO RANDOM STEER INPUT

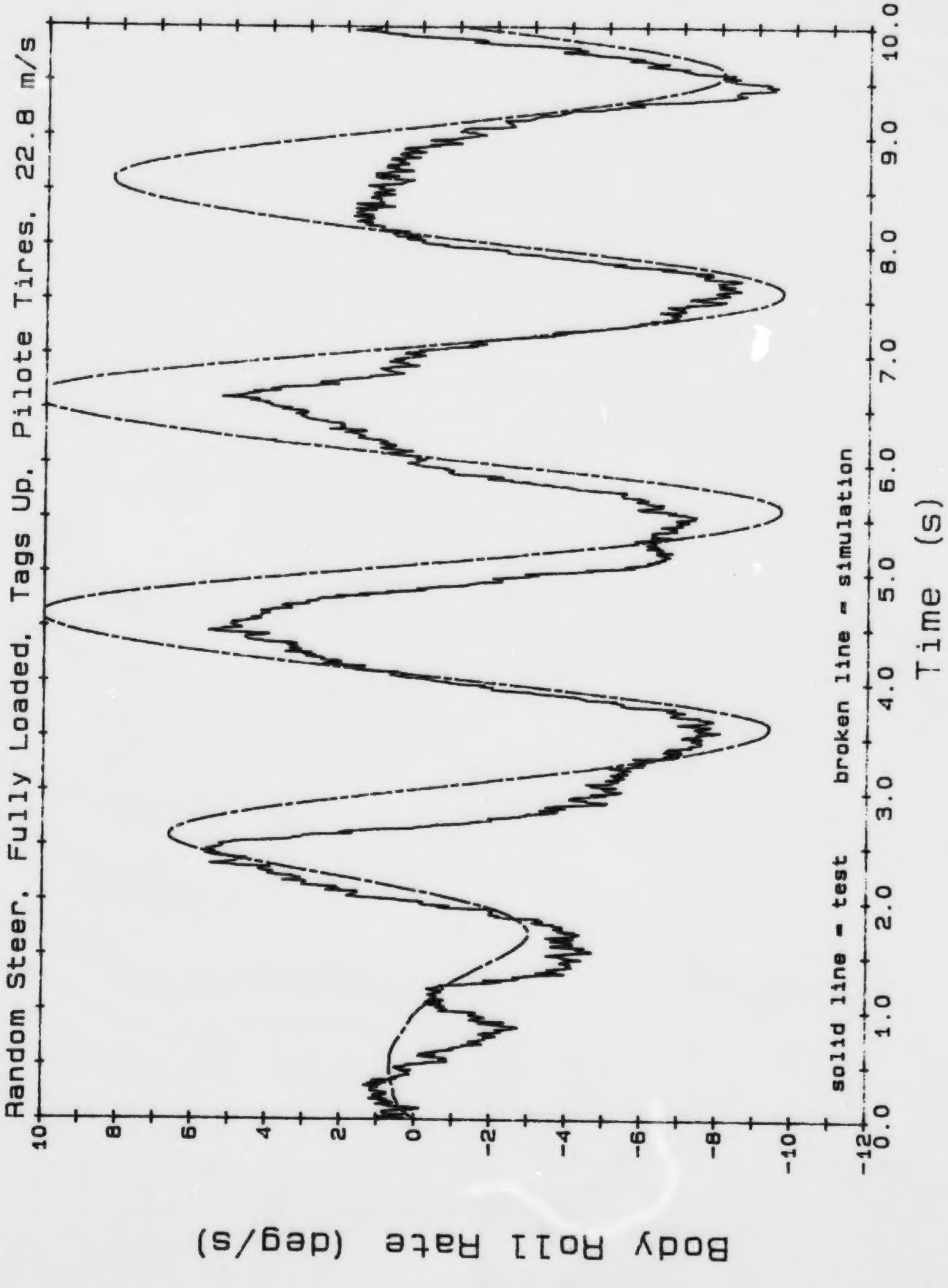


FIGURE 38: BODY ROLL RATE RESPONSE TO RANDOM STEER INPUT

6.0 ASSESSMENT OF ADVANCED SUSPENSIONS

6.1 SELECTED SUSPENSIONS

The review of the state-of-the-art in suspension design carried out during the program identified a number of promising alternatives which were anticipated would provide improvements in ride and perhaps handling when assessed on a qualitative basis. These configurations are described in Table 5. Configuration 1D is in fact the suspension selected for the advanced double deck bus as described in Reference 1.

Configuration	Description
1	Baseline Test Bus with front beam axle, Drive 'H' frame/beam axle; independent trunnion mounted tag wheels.
1A	Same as #1 except front beam axle is replaced by an independent MacPherson Strut airspring suspension.
1B	Same as #1 except Drive axle suspension is replaced by a rigid "A-frame" airspring suspension. Air springs act at an effective width much wider than the standard beam axle.
1C	A combination of #1A and #1B. Front independent MacPherson struts; Drive A-frame; and independent trunnion mounted tag wheels.
1D	Front independent MacPherson struts; pusher wheels mounted on MacPherson struts (Same as front axle) at the position that the drive axle was previously stationed; drive axle A-frame mounted at the position that the tag wheels previously occupied.

TABLE 5: ADVANCED SUSPENSION CONFIGURATIONS

Stiffness and damping at each suspension station for all five models are maintained effectively the same. Therefore changes in suspension performance between models are principally attributed to geometric changes only.

6.2 ANALYSIS OF SIMULATION RESULTS

The effect of each modification on bus ride and handling performance was examined using the simulation modules. Each bus model was subjected to the same vertical and steering input, and a comparison made of the response results.

6.2.1 Vertical Ride Response

To compare the vertical response characteristics, each bus model was simulated to proceed at a speed of 20 m/s (45 mph) over a half-sine "bump", 25.4 mm (1") in height and 1.5 m in half wavelength, with only the right side tires encountering the bump. This type of input qualifies as only one of several tests in evaluating the overall vehicle ride behaviour, and the results form only part of an overall examination.

In Figure 39, the vertical acceleration experienced at the bus body c.g. is shown for each model. It can be seen that modest improvements are made as the configuration is changed, particularly for model 1D where the "A-frame" drive axle is incorporated with a pusher style tag axle and front independent suspension. This mainly arises since the vehicle is softer in pitch, although the drive axle appears to be underdamped in the example shown. Body vertical acceleration, 1.3 m to the right of the c.g., is shown in Figure 40 to demonstrate the effects of body roll.

6.2.2 Handling Response

To examine handling performance, each bus model was given a steering step input, in which the front wheel angles were increased linearly from 0 to +10.5° (right turn) in a time of 0.75 seconds, and held at 10.5°.

The lateral acceleration response and yaw rate of the bus body is similar in each case and is now shown. However, the roll angle encountered by each case varies with each modification as shown in Figure 41. The suspension configuration of the test bus shows the greatest roll angles during the maneuver. Use of an independent front suspension alone shows some improvement with a decrease in peak roll angle of 13%. The use of a rear drive-axle A-frame either with or without the independent front suspension in configurations 1C, 1D and 1B shows a much greater reduction from the test bus suspension of order 36%.

6.2.3 Summary of Simulation Results

The simulation results show that the A-frame rear suspension contributes the most to an improvement in ride and handling when compared to the test bus. The independent front suspension is less successful in improving ride but this is partly a result of not softening the front spring rate to a different value giving a different compromise between ride and handling.

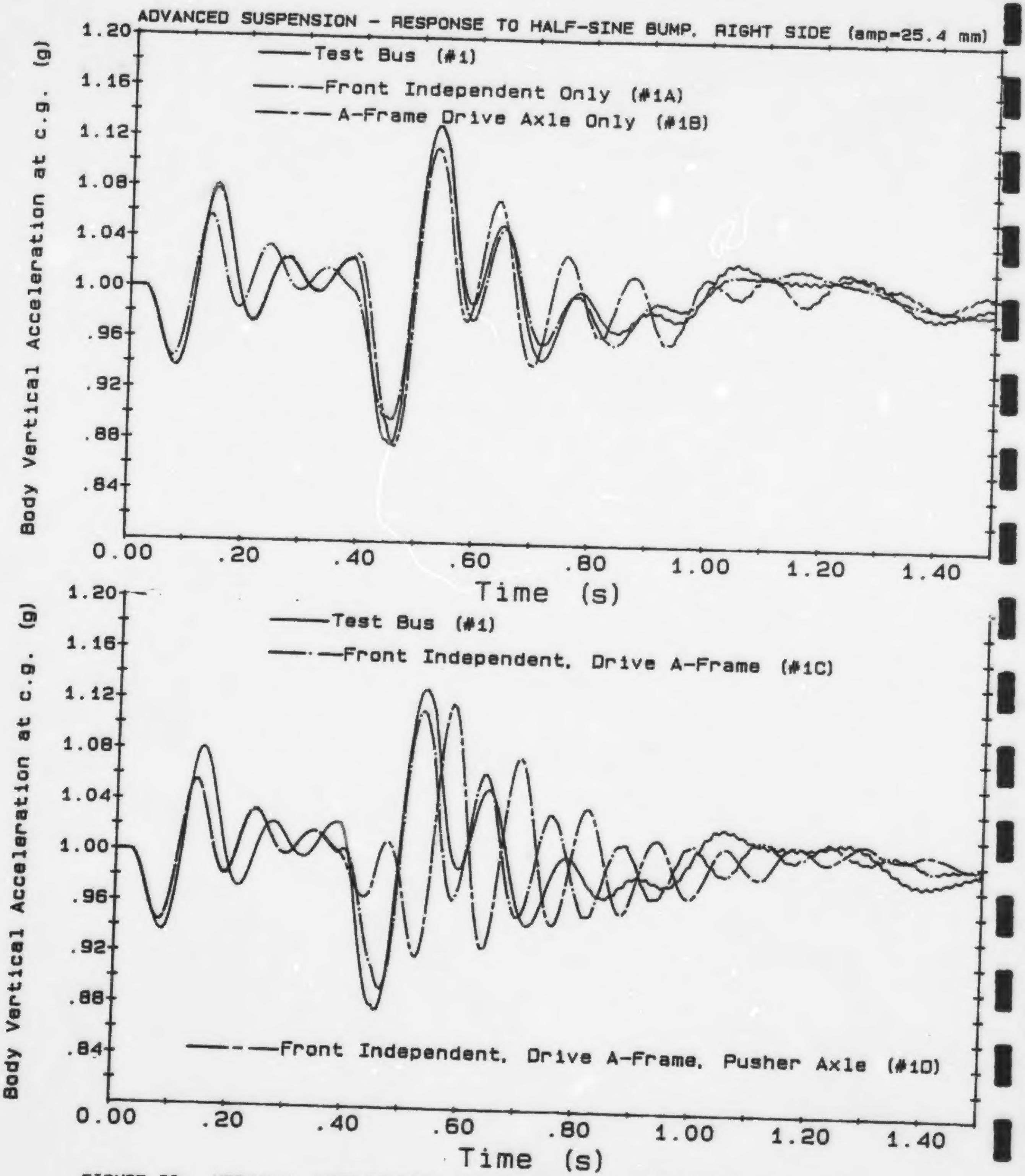


FIGURE 39: VERTICAL ACCELERATION OF BUS BODY FOR DIFFERENT SUSPENSIONS

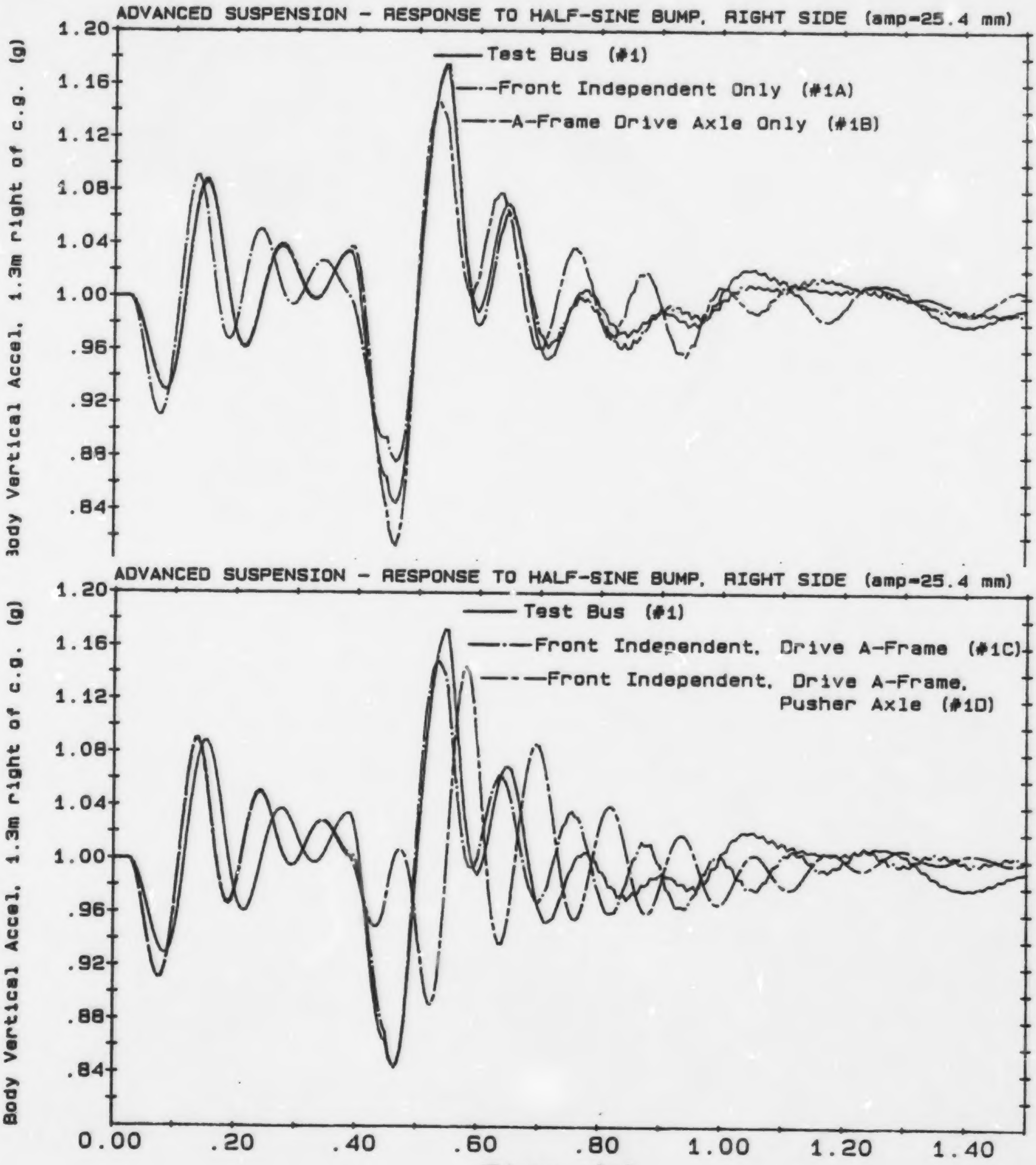


FIGURE 40: BODY VERTICAL ACCELERATION 1.3 M TO RIGHT OF C OF 6
FOR VARIOUS SUSPENSIONS

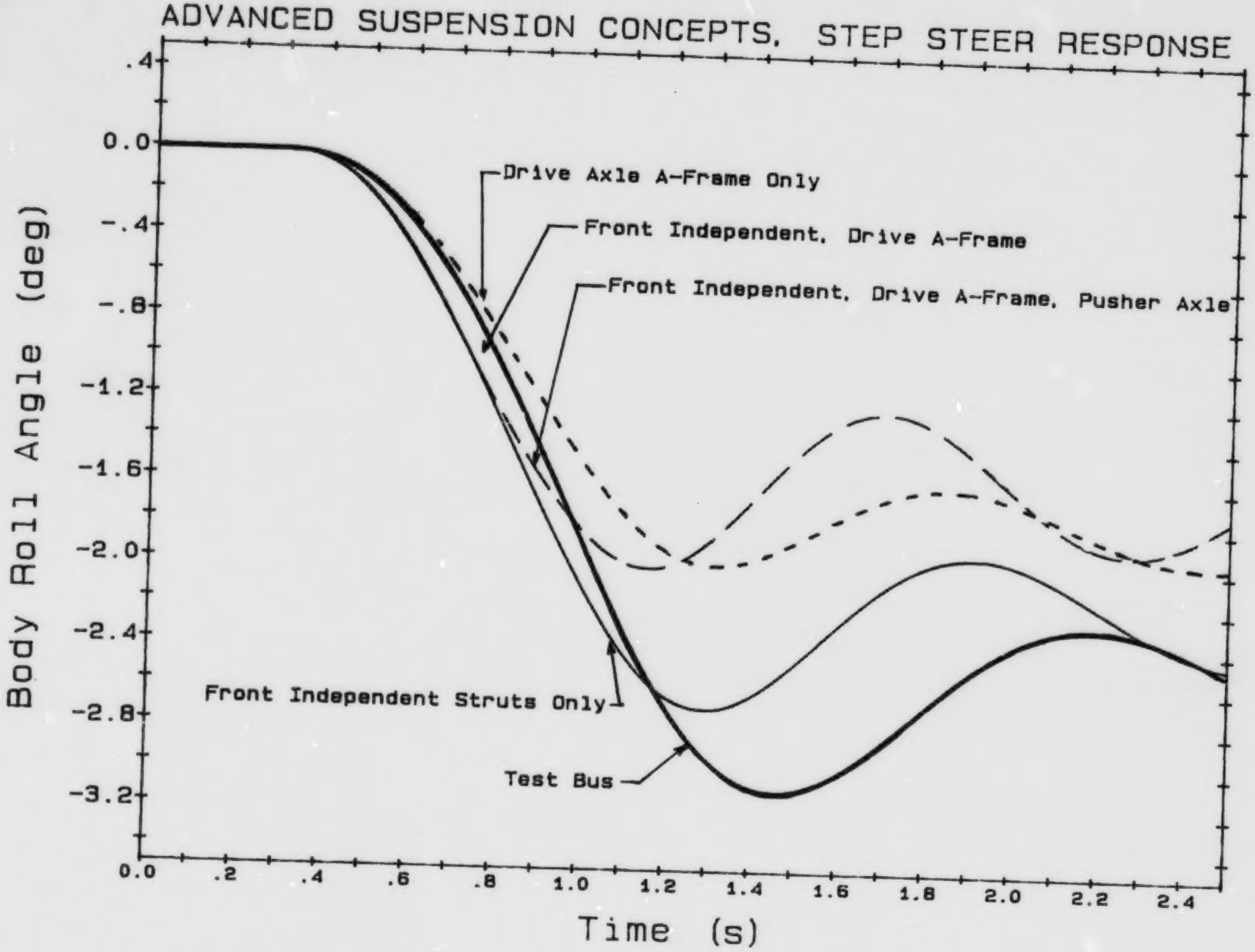


FIGURE 41: COMPARISON OF BODY ROLL ANGLE FOR DIFFERENT SUSPENSION SYSTEMS

7.0 ASSESSMENT OF ACTIVE SUSPENSIONS

7.1 STATE-OF-THE-ART REVIEW OF ACTIVE SUSPENSIONS

7.1.1 Terminology

Active suspension systems can take various forms for automotive use and therefore it is first necessary to define the terms used to describe various options and alternatives.

- **Adaptive Systems**

Systems which change fairly rapidly (up to 3 or 4 times/second) or slowly to vehicle or external inputs, e.g., damping change with speed or wheel stroke. (This could also include dynamic pitch and roll control).

- **Fully Active**

A system which replaces the conventional suspension components with a control system such as a hydraulic strut enabling full suspension modification to various vehicle and external inputs, e.g., the "Lotus" system. No system is yet in production. Several system manufacturers have experimented with these systems over the years. The major problem is not the hardware but the software control.

- **Partially Active**

A fully active component combined with passive elements. (e.g., a fully active damper combined with a passive spring).

- **Levelled**

A device that connects to vehicle trim height usually with change of load and often fitted to the reer only of cars, but usually fitted all round on buses.

It has been argued (15) that an active levelling system can improve overall system performance significantly.

- **Passive**

A system that has passive components only, with no external control.

7.1.2 Review of Progress in the Field

- **Levelling**

Levelling has been widely used on both cars and coaches for many years and is well understood. Most systems use an external power source either engine or electric motor driven using compressed air or hydraulics. Some car systems are "self pumping" using the energy from road bump inputs to "pump up" the vehicle by hydraulically pressurizing the damper. (A sealed low pressure twin tube design).

Karpoff shows (16) that a "high gain" (fast) load levelling device associated with active damping can give significant improvement in ride. He does not define "fast" but response time would have to be of the order of 10 msecs suggesting significant power consumption or energy storage using accumulators.

This type of system could be of interest for coaches but it is likely that to achieve the response times required, a hydraulic rather than pneumatic system would be needed, suggesting major redesign of the existing suspension systems.

o Adaptive Systems

Several luxury European and luxury and down market Japanese car manufacturers fit adaptive dampers which have either pneumatic or electric motor control. These have usually 3 different levels of damping by having several bleed orifices which are opened or closed by a rotating disc.

Adjustment steps are usually
high damping - no orifice
medium damping - 1 orifice
low damping - 2 orifices.

The following control signals are used in various systems.

Vehicle speed - above a certain speed dampers change to a higher damping.

Handwheel angle - above a certain handwheel angle dampers change to a higher setting (sometimes also associated with a minimum speed)

Wheel stroke - above a certain wheel stroke damping is increased (this is associated with the faster reacting systems.

Radar - one Japanese manufacturers uses a radar system that "sees" the road in front of the front fender. Although there is potential in this type of system (16) - particularly associated with fully active systems, much development is still required.

This type of adaptive system could be of interest for coaches. The technology exists but hardware of a size and energy absorbtion characteristics for a coach would have to be developed. The performance gains are limited but certainly some gain in comfort would result from a well developed system. It is unfortunate that the development potential has not yet been realized on some of the cars using this system.

Armstrong have developed a system termed ARC (adaptive suspension control) which is of this type but has a rather fast response time of the order of 30 msecs. and the manufacturer's claim a response that takes less than 1 metre of road at 80 km/hr. Potentially this can be used to produce some intermediate performance advantages between adaptive systems and systems with fully active dampers. It is being tried by several car manufacturers at present. With all adaptive and active systems the major problem for improved performance is optimizing the control system and its associated input requirements.

- **Partially Active**

Partially active systems are under development by several suppliers and automotive manufacturers. The desire to improve ride and the ride handling compromise are there.

Several solutions have been suggested. The most promising seem to be a fully active damper combined with a passive spring, with or without levelling. With active dampers Kawagoe & Iguchi (16) show that a 15% reduction in vertical acceleration and 18% reduction in pitch acceleration can be achieved and using the front wheel displacement as a preview control for the rear damper a 55% reduction in vertical acceleration and 65% reduction in pitch acceleration can be achieved up to 90 km/hr. On a coach with a wheelbase of 6m, the equivalent speed would be 216 km/hr. This preview control would give these significant performance gains under all normal operating speeds. As with all reactive systems the problems of measurement and control exist.

This category also includes handling oriented systems such as rear and "all-wheel" steer effects with lateral acceleration, speed, hand wheel angle, and yaw rate. These systems are technically interesting and can theoretically be used to improve and modify axle force steer effects. At present some passive systems perform at least as well. This status will probably change with development.

- **Active**

At present these systems are limited to research and development vehicles for road use. However, some special purpose military vehicles already use some features of these systems.

The best known of proposals put forward is the "Lotus" system. This solution originally put forward by Peter Wright of Lotus with control theory and computer control by David Williams from the Flight Systems and measurement department at Cranfield College of Advanced Aeronautics, Bedford, England. It is now being developed by a joint company set up between Moog (hardware), Lotus (engineering expertise) and General Motors (parent company of Lotus). Some 7 or 8 consultancy agreements, with automotive manufacturers have been signed and some 14 cars are running at present. References 17, 18, 19, 20 and 21 describe the system as it exists at present.

The hardware of hydraulic jacks (double acting) with MOOG electro-hydraulic servo valves is fairly straight-forward although valve performance in terms of frequency response and phase lag has to be improved. Cost at present is very high but MOOG and G.M. may well solve the high volume productionization problems. (Initially Far East production is envisaged to reduce labour costs). The major breakthrough is the control system which has been developed at Cranfield by the "fly by wire" experts with a lot of previous experience with variable stability aircraft.

At present up to 40 transducers are used as shown in Table 6.

Variables	Number of Transducers
Wheel displacement	4
Force into body	4
Unsprung mass acceleration	4
Wheel to body velocity	4
Body vertical acceleration	1
Body lateral acceleration	1
Body before and after acceleration	1
Yaw rate	1
Roll rate	1
Pitch rate	1
Speed	1
Handwheel angle	1
Pump flow	1
Pump pressure	1

TABLE 6: TYPICAL TRANSDUCER REQUIREMENTS FOR AN ACTIVE SUSPENSION SYSTEM

A 16 bit computer is used to minimize force into the body from the suspension and this is modified by wheel stroke and velocity. In addition roll, pitch and heave are controlled together with front/rear "roll stiffness" ratio for handling and a "track seeking" algorithm added to aid the driver to follow a demanding track. This feature is one of the most impressive advances achieved by the suspension system. Ride improvements at low frequency are impressive but pot holes, road joints, and road noise are less than impressive at present.

The system at present is an excellent development tool for practically modelling passive systems for test both rapidly and accurately to arrive at the optimum passive system during new vehicle development. Productionization of the hardware, transducers and control system is more difficult.

7.2 SELECTED TECHNOLOGY FOR MODEL EVALUATION

A practically achievable example of adaptive suspension is the programmed ride control (PRC) system patented by Ford Motor Company (Ref. 22). The system uses low shock absorber damping in normal driving for soft ride characteristics, and automatically switches to firm damping when required for improved handling. Hardware includes adjustable, fast responding shock absorbers in which hard damping is tuned by a conventional piston assembly while soft damping is accomplished by opening a rotary shutter valve to allow oil to bypass the piston valving. Actuation is by fast acting, bi-stable rotary solenoids, with typical response times of 30 milliseconds.

The control strategy for this system is based on "predicted" vehicle dynamic attitude changes arising from sensed driver control inputs. For example, sensors are used to measure the steering handwheel angle and vehicle speed, and a microprocessor estimates the lateral acceleration which should result. In this manner, action can be taken during the time lag between driver control input and vehicle response, whereas, had the lateral acceleration been measured, this response time would already have passed.

The PRC module monitors the input sensors and increases the shock absorber damping by a factor of 3:1 whenever the lateral acceleration is greater than 0.35 g, or whenever the vehicle speed is greater than 37 m/s (83 mph). The effect of this adaptive control strategy on the handling performance of the highway bus was examined using the simulation module BUSHAN.

7.3 ANALYSIS OF SIMULATION RESULTS

An algorithm was introduced to the handling integration scheme to monitor the right front wheel angle at each time step. (The steering handwheel is not presently part of the suspension model.) An approximate relationship was used to calculate the anticipated lateral acceleration as:

$$a_1 = \frac{U^2\ 6}{WB}$$

where U = forward speed (m/s)
WB = approximate wheel base of bus
S = front wheel angle (radians)

It is emphasized that, although the current lateral acceleration is already calculated at each time step, the control strategy uses an anticipated lateral acceleration based on the current wheel angle. When a_1 3.43 m/s^2 (0.35 g), the damping force through each shock absorber is increased to three times its usual value in a time corresponding to 60 milliseconds.

This control algorithm was used with the Test Bus model subjected to a steering step input to the front wheels. Comparisons were made between the handling response curves with and without the adaptive damping introduced. Only small differences were observed in the body yaw rate and lateral acceleration response. However, the response of the bus in roll is improved as shown in Figure 42 when the adaptive damping control is used. Both the peak response time and roll angle are reduced for this severe step input. The peak roll angle is reduced by approximately 17%.

The full potential of this adaptive damping would be realized in the suspension design by softening the shock absorbers to provide the best possible ride with only acceptable handling. Firm valving would then be provided to give the best possible handling with only acceptable ride when severe conditions are encountered.

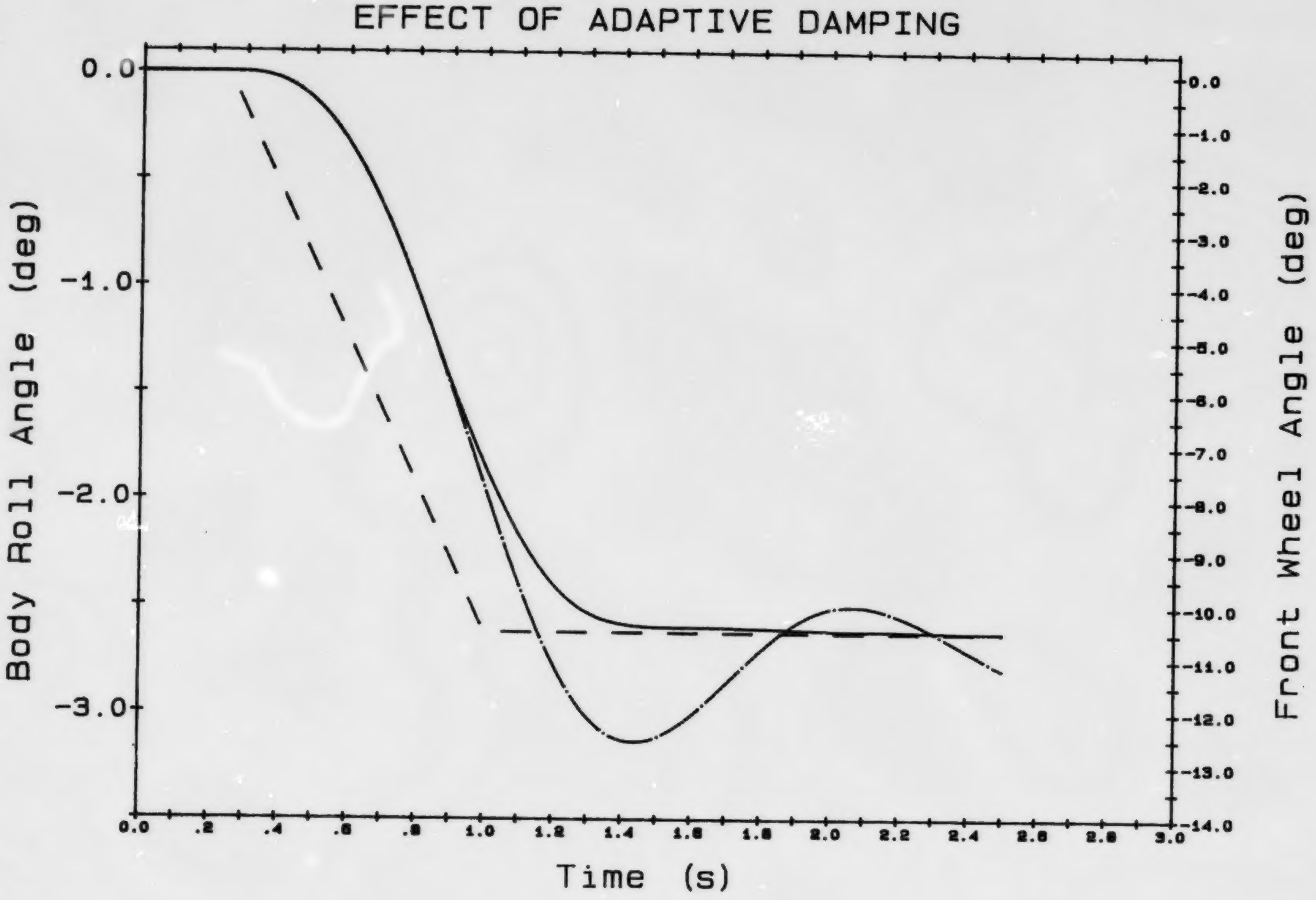


FIGURE 42: COMPARISON OF ADAPTIVE TO PASSIVE DAMPING

8.0 FORWARD PLANNING

8.1 DEVELOPMENT OBJECTIVES

The objectives are to develop a new suspension design which provides significant improvements in ride and handling of an inter-city bus while retaining the traditional virtues of rugged, low-cost, low-maintenance design typical of North American practice.

The assessment of suspension technology carried out, both qualitative and quantitative has resulted in the conclusion that the preferred configuration would be a relatively modest "incremental" style development rather than a radical new departure, such as fully active suspension and that significant improvements are still available with this approach in a cost-effective manner.

8.2 PROGRAM OPTIONS

The benefits of the more advanced techniques are unlikely to be achieved without radical re-design of the entire vehicle as a system. For example, the option considered of independent front suspension is unlikely to achieve its full potential unless bus body torsional frequency is above approximately 15 Hz which would require some structural changes to standard 40' inter-city buses. Another development in this category is the hydro-pneumatic suspension which only shows sufficient benefit to ride performance to justify development if a multi-axle configuration with longitudinal inter-connection is used. The A-frame rear suspension which has significantly improved roll stiffness and roll damping is also unlikely to be readily installed in current North American 40' buses.

Based on the above considerations, the following program options are considered viable and are segregated into near and long term solutions.

8.2.1 Near Term Solution

In the near term, it is considered that an H-frame rear suspension with conventional airbags, combined with, some form of adaptive damping would provide improved ride and vehicle handling. This program could be a relatively low-cost development and would be applicable to vehicles of the current type without major modification. A program of this nature would take approximately 15 months as shown in Figure 43 and would cost approximately $500,000.00 if a basic vehicle was made available for modification by a manufacturer for prototype test.

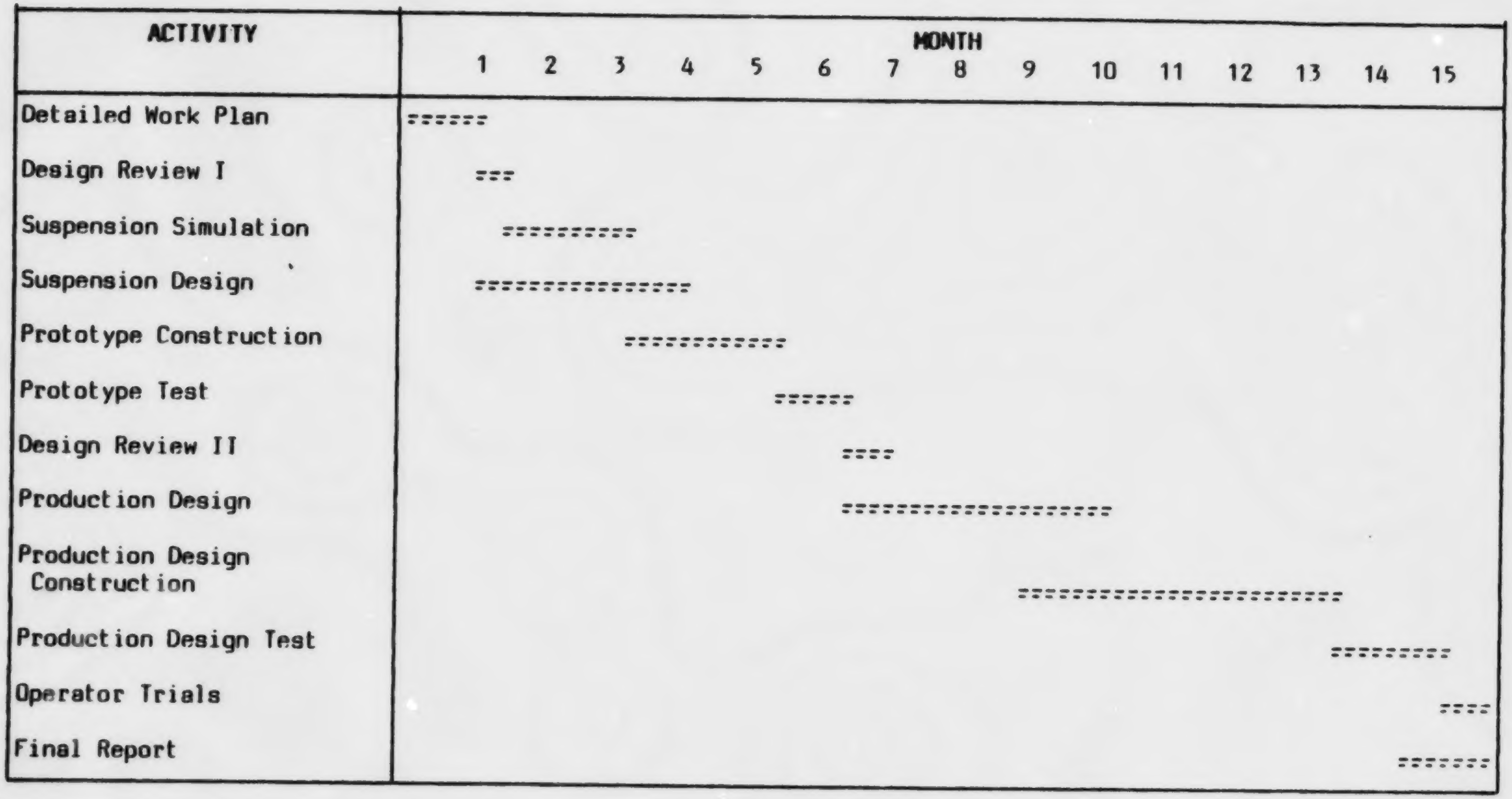


FIGURE 43: SCHEDULE FOR NEAR TERM DEVELOPMENT PROGRAM

8.2.2 Long Term Development

In the event that a radically re-configured vehicle were to be developed, for example, a 45 foot 4-axle or high capacity (double deck) configuration, then the benefits of a more advanced design could be realized, and in fact would be needed for either configuration. In this longer term program, the preferred configuration would utilize independent front suspension and the A-frame rear suspension as previously modelled and as recommended for the double deck bus. If 4 axles are needed for axle-load considerations, the hydro-pneumatic suspension is strongly recommended for development.

This longer term program would necessitate more extensive modelling and some level of component testing of the hydro-pneumatic units. This program is shown in Figure 44.

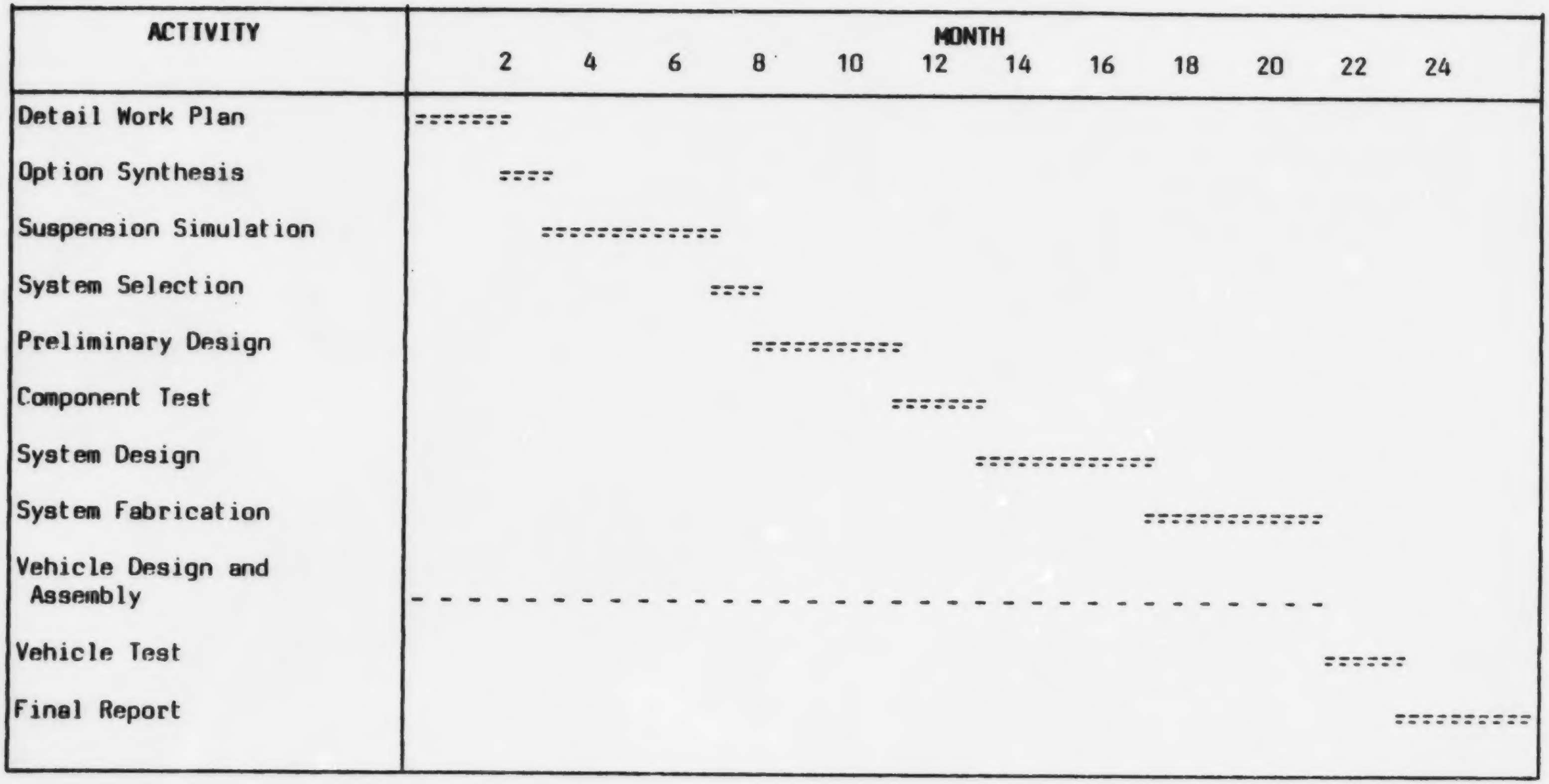


FIGURE 44: SCHEDULE FOR LONG TERM DEVELOPMENT PROGRAM

9.0 CONCLUSIONS AND RECOMMENDATIONS

9.1 CONCLUSIONS

It was concluded from the work that:

- current North American buses have rugged and long-life suspensions to suit road conditions, but improvements to ride and handling are both desirable and practical.
- Experience in Europe indicates that significant improvement is possible with relatively conventional incremental type developments.
- A fully active suspension is unlikely to be a cost effective solution in the near or medium term for intercity buses, and at the current stage of development there are serious reservations about reliability; maintainability and safety.
- The computer simulation technique developed agrees reasonably well with road test data for the MCI 102-43 test bus in both on and off design conditions, but the range of variation in parameters over which it was tested produced smaller effects than was expected.
- There is evidence to suggest that body stiffness will need to be modelled for a more accurate representation of real vehicle behaviour.
- The most significant undercar component with a strong effect on simulation results were the shock absorbers. It is also significant to note that there was a marked and undesirable variation in these components on the test bus.
- For effective simulation the non-linearity of the shock absorber and coulomb-type damping must be modelled.
- Some form of active damping control is a low cost and feasible technique for improving handling in manoeuvering while not causing a deterioration in ride in normal operation.
- Based on the modelling results of advanced suspensions the most promising layout for an advanced bus is with independent front suspension, pusher type tag axle, and A-frame style drive axle suspension.
- Large improvements in suspension performance are unlikely without major bus redesign.
- Some worthwhile improvements may still be made by increasing roll stiffness and damping by incorporation of an A-frame or wider based H-frame suspension on existing bus designs.

9.2 RECOMMENDATIONS

It is recommended that:

- Future modelling of buses incorporate at least some representation of body stiffness. (The developed program is capable of this.)
- A near term program focus on increasing roll stiffness and damping at the rear of the vehicle.
- A longer term program focus on independent front suspension with an A-frame drive axle and pusher type tag axle.

REFERENCES

1. J.H. Stannard, UTDC Research and Development Ltd., "The Conceptual Design of an Advanced Canadian High Capacity Highway Coach", Transport Canada TP 6300E, March 1985.

2. DSS Contract Serial No. ISD85-00205 "Assessment of Advanced Bus Suspension Technology", November 13, 1985.

3. "Intercity Coach Suspension Design; A State of the Art Review", TTL Report No. 449-510-001, February 1986.

4. International Standard I.S.O. 2631, "Guide for the Evaluation of Human Exposure to Whole Body Vibration", Second Edition, 1978.

5. L.B. Walker, Hovey & Associates ltd., "Selection Criteria for Heavy Duty Road Vehicle Suspensions", Transport Canada TP 4122, November 1982.

6. B.R. Davis, F.J.M. Salaborn, "Active Suspensions with Vibration Absorbers and Optimal Output Feedback Control, SAE 841253, 1984.

7. Technical Note on Citroen BX, Automotive Engineering, Volume 90 No. 12, P. 35, Dec. 1982.

8. "Rapid Suspension Changes Improve Ride Quality", Automotive Engineering, Mar. 1984.

9. A.E. Moulton & A. Best, "Hydragas Suspension", SAE 790374, 1979.

10. M.B. Packer, "Active Ride Control - A Logical Step from Static Vehicle Attitude Control", SAE 780050, 1978.

11. "FRP Halves Spring Weight, Improves Ride", Automotive Engineering, May 1981.

12. "Road Vehicles - Steady State Circular Test Procedures", I.S.O. Report 4138.

13. "Road Vehicles - Lateral Transient Response Test Procedures", I.S.O. Proposal 7401, Sent 1984.

14. Anderson, R.J., "A General, Mult-Body Approach to Dynamic Modelling of Rail Vehicles". Proceedings of the 3-rd International ASME Computers in Engineering Conference. Chicago, U.S.A., 1983, Vol. 1, pp 223-228.

REFERENCES (cont'd)

15. Dean, Karpoff, "Active Damping in Road Vehicle Suspension Systems", Vehicle System Dynamics 12 (1983), pages 291-316.

16. K. Kawagoe, M. Iguchi, "Semi-active Control and Optimum Preview Control Applications to Vehicle Suspensions", JSAE Review August 85, pages 24 - .

17. Automative Engineers, "Computer Control Suspension (C.C.S.)", August 1986, page 19.

18. D. Williams, "Active Suspension for Automobiles - A Further Progress Report", Cranfield Internal News Review (Unpublished).

19. J. Sinek, "Suspension Without Springs", Motors, 10 September 1983.

20. Alan Baho, "Lotus Active Suspension", Automotive Engines, Feb/March 1984.

21. P. Linert, "Tommorows Land Suspension", Road & Track, October 1986.

22. 1987 Thunderbid Turbo Coupe Programmed Ride Control (PRC) Suspension, Michael W. Soltie, SAE 870 540, Feb. 1987.

Volume

2

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VOLUME 2: APPENDICES

BY

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17. Abstract

The objective of this study was to quantify the benefits of advanced suspension systems when compared to existing North American suspension systems on highway buses. Existing computer simulation modules were modified where necessary and integrated into an overall program capable of simulating vehicle ride and handling in considerable detail at the component level. The simulation program was verified by a test program on an existing MCI 102-A3 intercity bus from which a variety of test results for steady state and transient handling and ride data were accumulated. The successfully verified model was used to assess a number of advanced suspension systems recommended for investigation as a result of a state-of-the-art survey carried out during the study.

Increased roll stiffness and damping is recommended for near-term development. It was concluded that the level of improvement possible with an independent front suspension and A-frame rear suspension was desirable but would require a longer term development as part of an advanced bus program.

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17 Résumé

La recherche dont il est rendu compte ici avait pour objectif de chiffrer les avantages que peuvent apporter des systèmes avancés de suspension pour autocars, par rapport aux systèmes existants en Amérique du Nord. Des modules de simulation informatique dûment adaptés ont été utilisés pour représenter à un niveau de détail très poussé la directibilité et la tenue de route des véhicules équipés des suspensions étudiées. Les résultats théoriques obtenus ont été validés par des essais routiers effectués avec un autocar MCI 102-A3 et visant à connaître les caractéristiques particulières de sa suspension en régime permanent et transitoire ainsi que sa tenue de route. Ainsi validé, le modèle de simulation a servi à caractériser un certain nombre de systèmes avancés de suspension qu'un tour d'horizon avait permis de recenser.

A court terme, il est préconisé d'accroître les raideurs en roulis et l'amortissement. La recherche a montré que les améliorations possibles grâce à une suspension avant à roues indépendantes et à une suspension arrière à cadre triangulaire sont souhaitables, mais que des travaux plus poussés sont nécessaires dans le cadre d'un programme d'étude d'un autocar de type avancé.

18. Mots-clés
Autocar de type avancé,
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ATTACHMENT A-1

TEST REPORT - SUSPENSION, MODERN NORTH AMERICAN HIGHWAY BUS

Report No.: 449-630-001

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SUMMARY

This report describes a series of tests carried out to examine the ride and handling performance of a typical highway bus. Objectives of the test were to provide measurements to validate a computer simulation of the bus suspension, gather baseline data on the present performance of highway bus suspension systems, and to measure vehicle and suspension component characteristics required as input to the computer simulation.

A modern highway bus, leased from a prominent bus manufacturer for a period of 3 months, was tested for inertial, suspension component, ride quality and handling characteristics. The tests were carried out with the bus in its normal production configuration and in various off-design conditions. Objectives, procedures and results are given for each test.

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1.0 INTRODUCTION

1.1 BACKGROUND

TTL has been contracted by the Transportation Development Centre (TDC) to study advanced suspension systems for highway buses. A general objective is to improve the ride and handling characteristics of current and future highway bus designs, thereby encouraging greater public use of intercity bus transportation.

To reach this objective, the development of a computer simulation is being undertaken in order to provide a verified and accurate tool for advanced bus suspension design. Use of the simulation will allow a preliminary evaluation of candidate suspensions with confidence and without resorting to expensive and time consuming prototype test programs.

The project objective is to develop, verify and use a vehicle suspension system computer simulation to make quantitative predictions of the benefits of advanced suspension systems applied to highway buses.

1.2 GENERAL TEST OBJECTIVES

The fundamental purpose of the test program was to provide data for the development and verification of a general suspension system computer simulation for the highway bus.

Particular objectives established at the outset of the test program (Test Plan, TTL Report No. 449-610-003) were to:

- instrument and test a modern highway bus for ride and handling performance; gather baseline data on the present performance of highway bus suspension systems for a range of operating conditions.

- measure vehicle and suspension component characteristics required as input to the computer simulation.

- modify the vehicle to known off-design conditions and re-test; gather sufficient performance data for comparison with computer predictions derived with corresponding conditions imposed.

2.0 METHOD OF APPROACH

The test approach focused on the principal objective to provide validation for the computer suspension simulation, specified in TTL Report 449-540-001 and presently in development. To place this objective in context, the simulation is intended as a design tool at the component level; that is, the designer inputs the geometric locations and connection points of each suspension component, it's force/deflection/velocity characteristics, and the inertial properties (centre of gravity, moments of inertia) of each rigid body being connected. For specified road and steering inputs, the simulation will predict the ride and handling performance of the vehicle system.

The test program was planned to examine the suspension of a particular highway bus which would, in parallel, be mathematically modelled for the computer simulation. Component locations, properties, etc. were measured to substantiate the simulation input, and the bus was instrumented to measure the ride and handling performance and to offer check points for the computer simulation.

During the performance testing period, practical parametric changes were made to the actual vehicle and suspension system (off-design conditions). Thus, comparison could be made between the actual performance results and those predicted when corresponding changes were made in the computer model. In this manner, a high level of confidence could be reached in the general computer simulation of bus dynamic performance for use with advanced suspension concepts.

2.1 TEST BUS

To carry out the tests, a modern highway bus was leased from a prominent bus manufacturer, Motor Coach Industries, for a period of three months from April 17 to July 16, 1986. The chosen bus, Model 102-A3, is representative of the contemporary highway bus industry in North America.

The particular bus, shown in Figure 2.1, was one of the first of the Model series, built in 1985 and used as an engineering prototype vehicle. It was identical to the production bus, but as yet had not been introduced into revenue service. Thus, somewhat more flexibility was allowed in the types of tests which could be carried out.

General characteristics of the bus are:

Length	12.2 m (40')
Width	2.6 m (102")
Height	3.4 m (134")
Gross Vehicle Weight	17,146 kg (37,800 lb)
Number of axles	3
Type of Axles	front, beam axle drive, beam axle, dual tires tags, independent
Type of Suspension	air springs, all axles

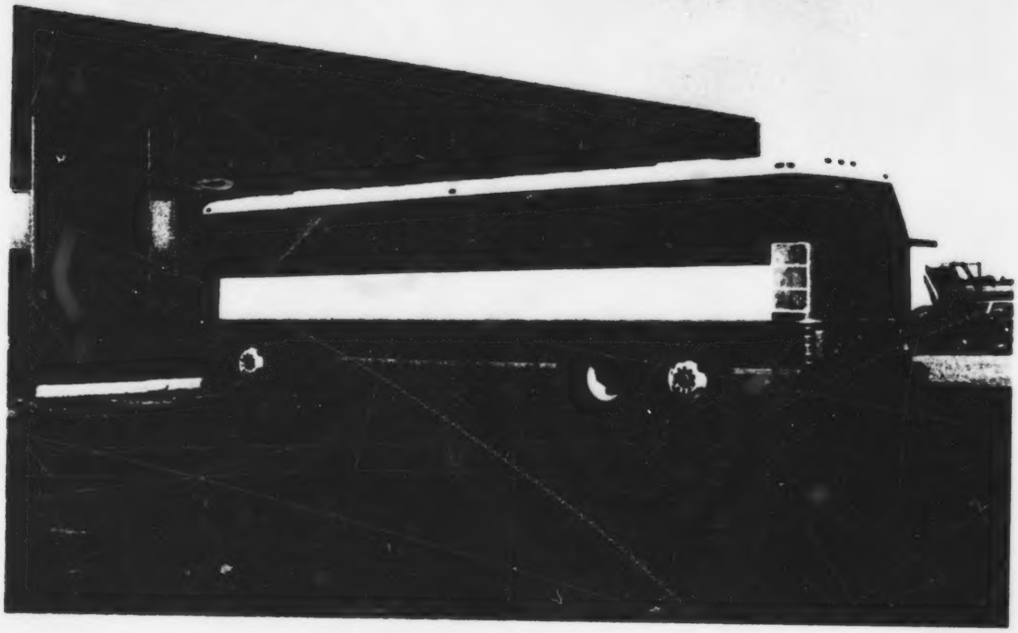


FIGURE 2.1: TEST BUS

To meet the objectives of the program, a test matrix was produced, as given in Table 2.1, encompassing tests of the bus in its present configuration, and tests in well defined, off-design conditions.

The off-design tests were, of course, limited to practical, non-destructive changes such that the bus could be returned in its original condition.

The various bus configurations are briefly described below.

Test Configuration #	Static	Dynamic					
		Ride Quality			Steady State Handling	Transient Handling	
	Weight/c.g.	Random Input	Step Input	Harmonic Input	Constant Yaw Rate Input	Random Steer Input	Step Input
1							
2	X						
3		X	X		X	X	X
4						X	X
5		X		X			
6				X			
7				X			
8	X	X			X	X	X
9					X	X	X
10	X					X	X
11	X				X	X	X
12		X					
13		X					
14		X					
15		X					
16		X					

TABLE 2.1: TEST MATRIX

CONFIGURATION #	DETAILS
1	Bus unloaded, standard tires (pressures 111 psi-front, 75 psi-drive, 65 psi-tag) (Condition as received from bus manufacturer), tags down.
2	Bus unloaded, standard tires (pressures 111 psi-front, 75 psi-drive, 65 psi-tag), tags chained-up.
3	Bus unloaded, standard tires (pressures 100 psi-front, 80 psi-drive, 70 psi-tag), tags down.
4	Bus unloaded, standard tires (pressures 100 psi-front, 80 psi-drive, 70 psi-tag), tags depressurized (not lifted).
5	Bus uniformly loaded, standard tires (pressures 100 psi-front, 80 psi-drive, 70 psi-tag), tags down.
6	Bus uniformly loaded, standard tires (pressures 100 psi-front, 80 psi-drive, 70 psi-tag), tags depressurized.
7	Bus uniformly loaded, standard tires (pressures 100 psi-front, 80 psi-drive, 70 psi-tag), tags depressurized, anti-roll bar disconnected.
8	Bus uniformly loaded, low profile tires (pressures - all at 100 psi), tags down.
9	Bus uniformly loaded, low profile tires (pressures - all at 100 psi), tags chained-up.

CONFIGURATION #	DETAILS
10	Weights shifted to front and back, low profile tires (pressure - all at 100 psi), tags down.
11	Weights shifted to front and back, low profile tires (pressures-all at 100 psi), tags chained-up.
12	Bus unloaded, low profile tires (pressures-all at 100 psi), tags chained-up.
13	Bus unloaded, low profile tires (pressures-all at 100 psi), tags chained up, drive shocks disconnected.
14	Bus unloaded, low profile tires (pressures-all at 100 psi), tags chained up, anti-roll bar disconnected.
15	Bus unloaded, low profile tires (pressures-all at 100 psi), tags pressurized, anti-roll bar disconnected.
16	Bus unloaded, low profile tires (pressures-all at 70 psi), tags pressurized.

A more detailed explanation of configurations is given in Appendix 1.

Note: Standard Tires = Michelin XZA 12R22.5
Low Profile Tires = Michelin Pilote XZA-1 275/80R

2.2 BASELINE TESTS

The baseline configuration is essentially defined by configuration #3; the bus was unloaded (the curb condition) with standard bus tires (Michelin XZA 12R22.5) having pressures set to 100 psi-front axle, 80 psi-drive axle, and 70 psi-tag axle. The air springs for the tag axles were fully pressurized (35 psig).

In the baseline condition, the following tests were performed:

1) Weight and centre of gravity location. These tests are described in Section 3.0. (Note: Weight and centre of gravity location were actually measured with the bus in Configuration #2; since neither tire pressures nor chaining of the tags have a significant effect upon the measurements, this is considered a baseline test.)

2) Ride Quality - Random Road Input.

   Random road input consisted of driving the bus along pre-selected portions of highway at different speeds and recording all relevant ride data as detailed in Section 5.0.

3) Steady-State Handling

   Steady State Handling tests were performed by driving on a constant radius circle at varying speeds and recording all relevant handling data. Section 6.0 describes the Steady State Handling tests.

4) Transient Handling

Transient Handling tests were composed of two types of steering inputs. The first was an input in the frequency domain called "Random Steer Input" which consisted of driving the bus at constant speed while steering left and right in a sinusoidal fashion (amplitude ≈ ±90° on the hand wheel) starting first at low frequencies (0.2 Hz) and then increasing gradually to higher frequencies (up to 2 Hz where possible). The intent was to achieve a reasonable steering frequency content over the range 0.2-2 Hz.

The second transient handling test incorporated a steering step input. The focus for this test was to measure a time domain response of the bus when subjected to a rapid steering step input. The technique here was to drive the bus at constant throttle in a straight line and, at the selected time, turn the handwheel, as rapidly as possible, to a particular position and hold there for a minimum of 3 seconds to allow a new steady state to be achieved.

Section 7.0 details the Transient Handling tests.

2.3 PARAMETRIC TESTS

The parametric tests involved the altering and removal of certain suspension components and repeating most of the previous static and dynamic tests. The parametric changes included the following:

1) Depressurizing tag axle air springs:
2) Chaining-up tag wheels;
3) Changing tire pressures;
4) Changing tire type;

5) Disconnecting anti-roll bar;
6) Disconnecting shock absorbers;
7) Adding weight; and
8) Shifting weight.

1) Depressurization of the tags was accomplished by opening two 3-way valves (a provision supplied by the bus manufacturer) which exhausted air to the atmosphere. This essentially reduced the air pressure in the tag air springs to zero (gauge pressure) leaving the tag wheels resting on the ground.

2) Chaining the tags up is an extension of depressurizing the tags. After the air was exhausted from the tag springs, the tag wheels were raised above ground level (using 'run-up blocks') and chained in place. This left approximately 10 mm of clearance between the bottom of the tire and the ground.

3) Tire pressures were changed primarily to give different ride characteristics. The "standard tires" (Michelin XZA 12R22.5) mounted on the bus were received at pressures of:

front - 111 psi
drive - 80 psi
tag - 65 psi

For the majority of tests with standard tires, pressures were set to:

front - 100 psi
drive - 80 psi
tag - 70 psi

The "Low Profile tires" (Michelin Pilot XZA-1 275/80R) were used primarily with all tire pressures set to 100 psi. (One Ride Quality Test was run with all tires set to 70 psi).

4) Two types of tires were used during the bus testing:

   a) Standard tires (Michelin XZA 12R22.5)
   b) Low Profile tires (Michelin Pilot XZA-1 275/80R)

   Also, Alcoa Aluminum rims were used with the Low Profile tires, to reduce the wheel weights.

5) The anti-roll bar was disconnected by removing the clamps which held it to the body. It was then held in place using a rope from the bus body to the centre of the anti-roll bar. This made it unnecessary to completely remove the anti-roll from the bus during testing.

6) To remove some primary damping effects during testing, the shock absorbers (in this case the two rear drive shocks) were completely removed by unscrewing the top (body) and bottom (axle) mounting nuts. One ride test was performed in this condition.

7) Another parametric change was the addition of weight to the bus body. 47-100 lb steel bars and 52-25 lb lead bricks were uniformly distributed inside the bus with the weight being divided between the seats and the floor in a ratio to simulate the weights a "typical" passenger would apply to the seat and floor. (11.4 kg (25 lb) on the floor and 56.8 kg (125 lb) on the seat was used in this test program).

8) Weight was shifted within the bus body by placing a portion of the dead weight at the back of the bus (over the drive axle) and the remaining portion close to the front of the bus. 50% of the weight was placed at the back approximately 3.3 m from the c.g. and the remaining 50% was placed approximately 3.3 m to the front of the c.g. This altered the yaw and pitch inertias of the bus body while unaffecting the centre of gravity.

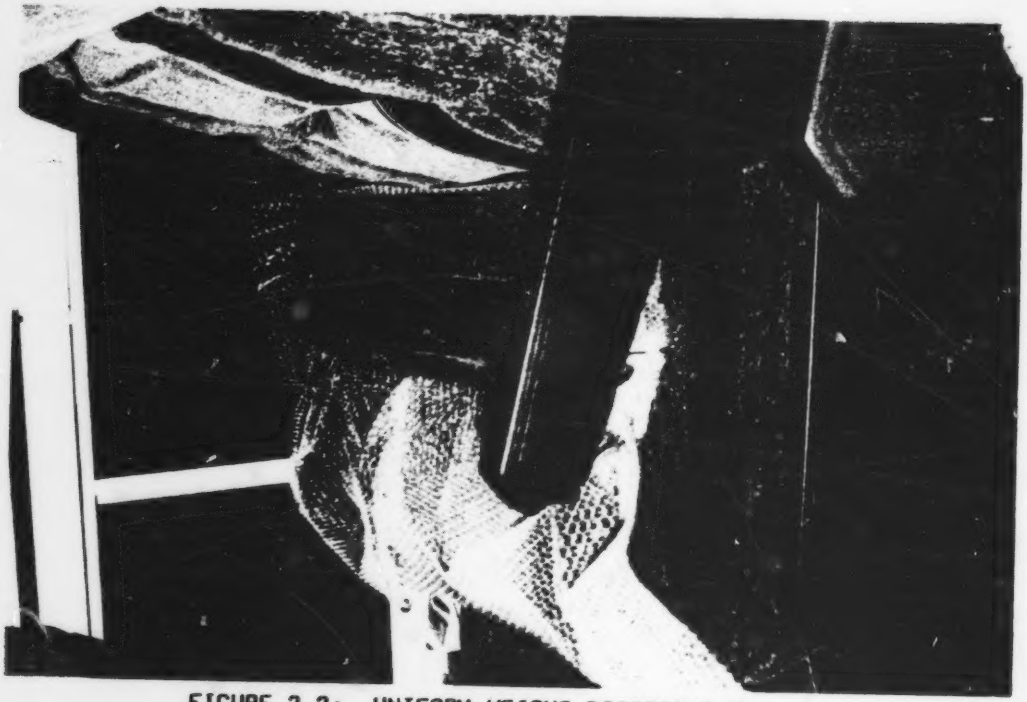


FIGURE 2.2: UNIFORM WEIGHT DISTRIBUTION

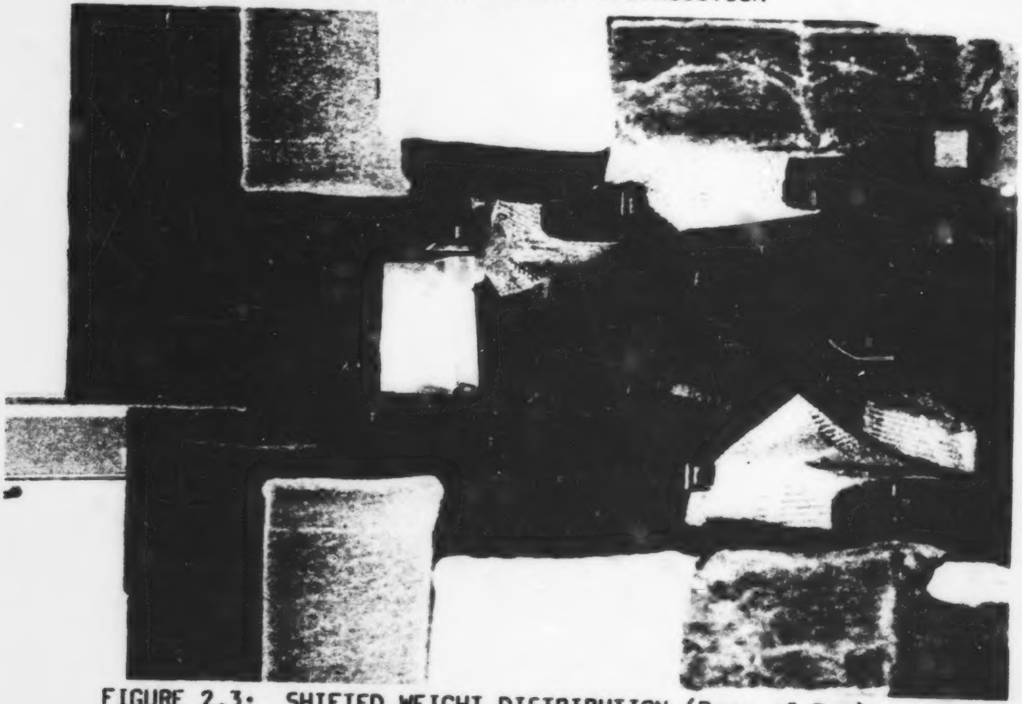


FIGURE 2.3: SHIFTED WEIGHT DISTRIBUTION (Rear of Bus)

One additional test, performed with the bus in several of the "off-baseline" configurations, was the "harmonic input" test.

The "harmonic input" test consisted of placing the bus on a four post "shaker", (2 tag wheels and 2 drive wheels were placed on the four posts) and cycling the posts over sinusoidal displacements. The frequencies and amplitudes of the motion of all four posts was the same; however, the phase relationship between posts was either in-phase or 180° out-of-phase, side-to-side. The frequency range covered from 0.1 Hz up to 11 Hz with amplitudes of ±1/8", ±1/4" and ±1/2".

3.0 VEHICLE INERTIAL PROPERTIES

3.1 TEST OBJECTIVES

The objectives of the inertial tests were to determine the relevant properties of the test bus and its configuration in order to substantiate input to the computer simulation. The properties determined through test include:

1) a) vehicle curb weight;
   b) body weight;
   c) front axle weight;
   d) drive axle weight; and
   e) tag axle weight.

2) a) vehicle centre of gravity locations.

3.2 TEST APPROACH

3.2.1 Mass Measurements

The relevant masses were determined by weighing the bus on a flat level concrete surface using four calibrated load cells. The load cells were situated to support the bus either by the axles or the body depending on the parameter being measured.

In addition, one set of mass measurements were made on a "drive-on" vehicle weigh scale at Blainville, Quebec.

3.2.2 Vehicle Centre of Gravity Locations

The location of the vehicle centre of gravity in a horizontal plane could be calculated from the mass measurements and support locations made in 3.2.1. The vertical position of the centre of gravity was not determined experimentally.

3.3 INSTRUMENTATION AND EQUIPMENT

The primary instrumentation and equipment implemented in the tests were the following: (not including the MCI model 102A3 highway bus)

3.3.1 Mass Measurement

a) Load-Weigh kit (4 load cell aircraft weighing kit) (TMF)

b) Vehicle Weigh Scale (Blainville, Quebec)
Model - Fairbanks Morse H90-7300
Capacity: 28,000 kg
Accuracy: 0.1%

3.3.2 Vehicle Centre of Gravity Locations

a) 4 load cell aircraft weighing kit (TMF).
b) Tape measure, accurate to ±.5 mm.

3.4 TEST SET-UP AND LOCATION

3.4.1 Mass Measurement

a) In preparation for the mass measurement of the test bus, it was necessary to ensure that the bus was in its "curb" condition. For this purpose, all luggage bays were emptied, all excess weight was removed from the interior of the bus, and all fluid levels filled. The bus was positioned in Bay #3 of the on-site Test and Maintenance Facility (TMF) on a flat, level section of concrete.

Weight measurements took place during the period May 5, 1986 to May 7, 1986.

b) The second set of mass measurements took place at the Transport Canada Motor Vehicle Test Centre in Blainville, Quebec on July 8, 1986 and July 9, 1986 with the bus fully loaded.

3.4.2 Vehicle Centre of Gravity Locations

The centre of gravity locations were determined as a 'by-product' of the mass measurement tests, by accurately measuring the location of the support points.

3.5 TEST PROCEDURES

3.5.1 Mass Measurement

a) The total vehicle weight and the overall centre of gravity location were determined by supporting the vehicle at four axle points (tags chained up). Clearances between each of the drive tires and the front tires were measured to ensure the vehicle was level. Load cell readings were then made. Also, positions of the four load cells were noted.

The sprung mass plus tag axles was measured with four load cells under four body points. The body was raised approximately 2 cm by operating the air levelling valves manually. With the body in this raised position, the load cells were positioned and "shimmed" such that, as air was slowly exhausted from the air springs, all body points made contact with the load cells at approximately the same time. Finally, all air was exhausted from the drive and front air springs (note: tags were chained up) such that the body was supported entirely by the load cells, and the axles (unsprung mass) were supported by the ground. Load cell readings were made.

The mass of the sprung mass plus tag axles and front axle was then measured. In this case, two load cells were placed under the front axle, and two load cells supported the body near the drive axle. This time, air was exhausted only from the drive axle air springs such that the weight of the drive axle was supported by the ground.

Again tags were chained up. Finally, load cell readings were made.

The mass of the sprung mass plus front axle was measured next. The procedure was exactly the same as with the previous case (sprung mass plus tags plus front axle) except that the tag axles were unchained but depressurized such that their weight was supported by the ground.

3.5.2 Vehicle Centre of Gravity Locations

Measurements of the sprung mass (bus body) and overall centre of gravity locations were performed simultaneously with the previous mass measurement (3.5.1) tests. The only additional data required was the location of the loadcells.

3.6 METHOD OF ANALYSIS

3.6.1 Vehicle Centre of Gravity Location

Only the centre of gravity location in a horizontal plane can be determined from the tests of 3.5.1 and 3.5.2.

These are determined by a moment balance using the data from the load cells.

3.7 RESULTS

The following masses were determined for the various components:

Vehicle Curb Weight	13,780 kg
Body Weight (sprung mass)	10,652 kg
Front Axle	916 kg
Drive Axle	1,806 kg
Tag Axles (combined weight)	406 kg

The body centre of gravity location in the horizontal plane was found to be:

500 mm behind the body geometric centre

22 mm to the right of the body geometric centre.

4.0 SUSPENSION COMPONENT TESTS

4.1 TEST OBJECTIVES

To test static and dynamic properties of selected components from the Test Coach as leased to TTL such that sufficient information is available to accurately model and substantiate the components' behaviour.

Specifically, the following components and properties were examined:

1) Shock Absorbers - damping properties.
2) Radius Rod - radial spring rates of the rubber bushings.
3) Air Spring - spring rates as well as interconnection effects.
4) Tires - non-rolling vertical spring rates.

4.2 TEST APPROACH

The approach taken was to remove key components from the bus suspension system. Generally, the component being studied was installed in the material test machine and tested dynamically for its force-deflection/velocity characteristics.

An appropriate amplitude of oscillation was determined for each component, which was then cycled manually at that amplitude quite slowly in order to view quasi-static characteristics. Next, the component was cycled sinusoidally starting at very low frequencies (0.1 Hz) and gradually increased until higher frequencies were reached (2.0 Hz - depending on the amplitude). At higher frequencies, the amplitudes were reduced since the test machine was limited in travel speed.

4.3 INSTRUMENTATION AND EQUIPMENT

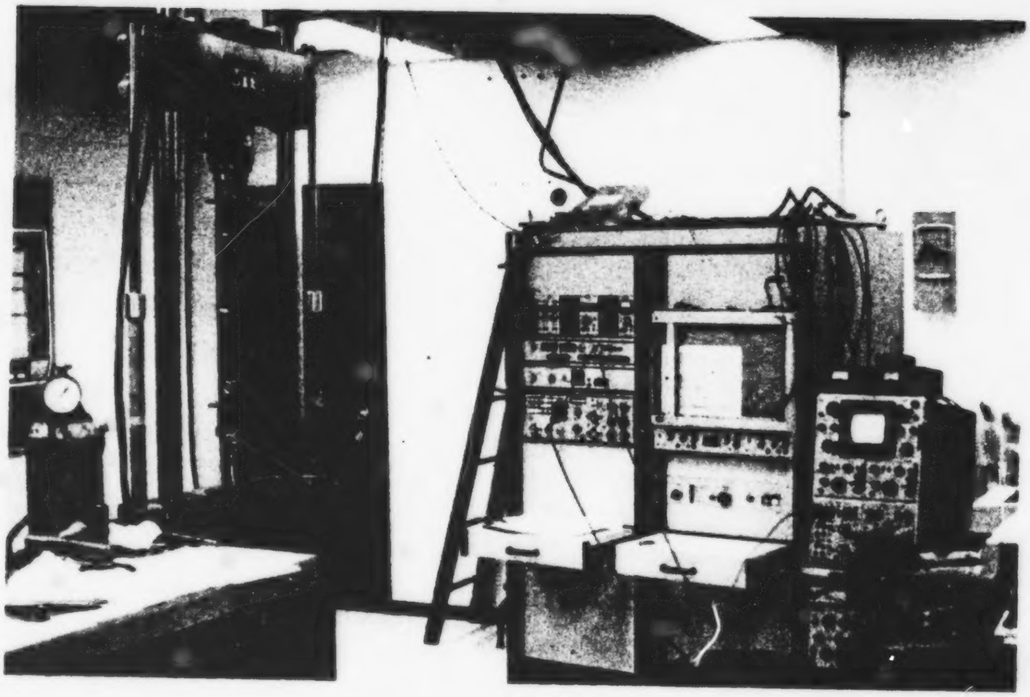
- Test Equipment:
  - MTS Model 810 Material Test Machine (50 Kip capacity);
  - Mounting brackets (manufactured by RMC) for tested components;
  - Hand-held 0-160 psig pressure gauge; (model P1530)
  - Torque wrench;
  - Tape measure; and
  - Digital thermometer (model HH-70TC).

- Test Components:
  - Shock Absorbers - 2 front axle shocks (Gabriel 690029)
    - 1 forward right drive axle shock (Gabriel 680211);
  - Radius Rod - 1 upper left front axle trailing link;
  - Air Spring - 1 right front axle air bag (Goodyear 1R12-635);
  - Tires - 1 Michelin XZA 12R22.5 (Standard Bus Tire)
    - 1 Michelin Pilot XZA-1 275/80R (Low Profile Truck Tire)

4.4 TEST SET-UP AND LOCATION

- Location: Royal Military College
  Kingston, Ontario
- Date: 21 May - 27 May 1986
- Personnel in attendance: Mark Guimont (MTS operator, RMC), Dr. C. Fortin (RMC), Jim Tregenza (TTL).

The test set-up is shown in Figure 4.1.



FIGIRE 4.1: TEST SET-UP -- INCLUDES MTS FUNCTION GENERATOR, CONTROLLER, PEN PLOTTER AND ACTUATOR

4.5 TEST PROCEDURE

4.5.1 Shock Absorbers

The mounting brackets for the shock absorbers were first secured to the MTS machine. A shock absorber was then mounted (with rubber bushings) and the nuts were torqued to 155 ft-lb as recommended in the MCI maintenance manual.

A nominal height for the shock absorbers of 543 mm was selected according to the static ride height of the bus and each shock absorber was cycled sinusoidally ± 12.7 mm about this nominal position. As well, each damper was cycled at nominal positions 25 mm above and below 543 mm to check for any length effects.

Each shock absorber was cycled at a range of frequencies varying from 0.1 Hz to 1.0 Hz (1.0 Hz @ ± 12.7 mm displacement was found to be the limit of the MTS machine - the maximum hydraulic flow rate (≈ 3.1 in/sec ram speed) is achieved under these conditions).

The shock absorber case temperature was observed during any particular test using the digital thermometer with a beaded thermocouple and approximate values were recorded. Finally, plots of force versus displacement were recorded on the MTS pen plotter.

4.5.2 Radius Rod

The radius rod mounting brackets were mounted onto each end of the radius rod using a rubber mallet. With the brackets approximately positioned, the assembly was mounted in the MTS testing machine, and mounting bolts were tightened. Next, the

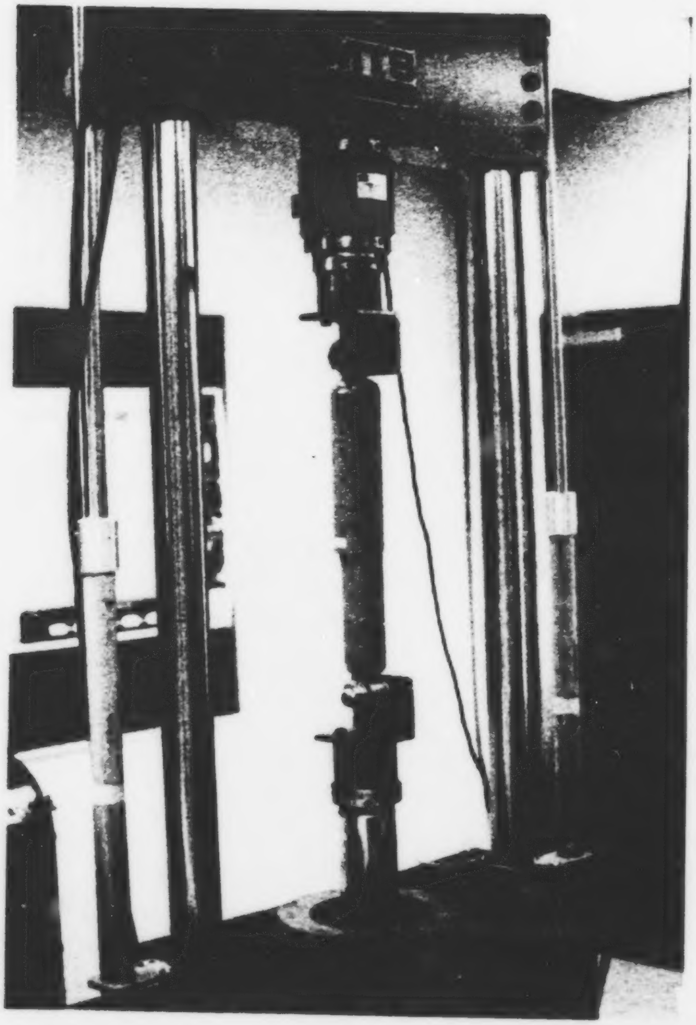


FIGURE 4.2: DRIVE SHOCK

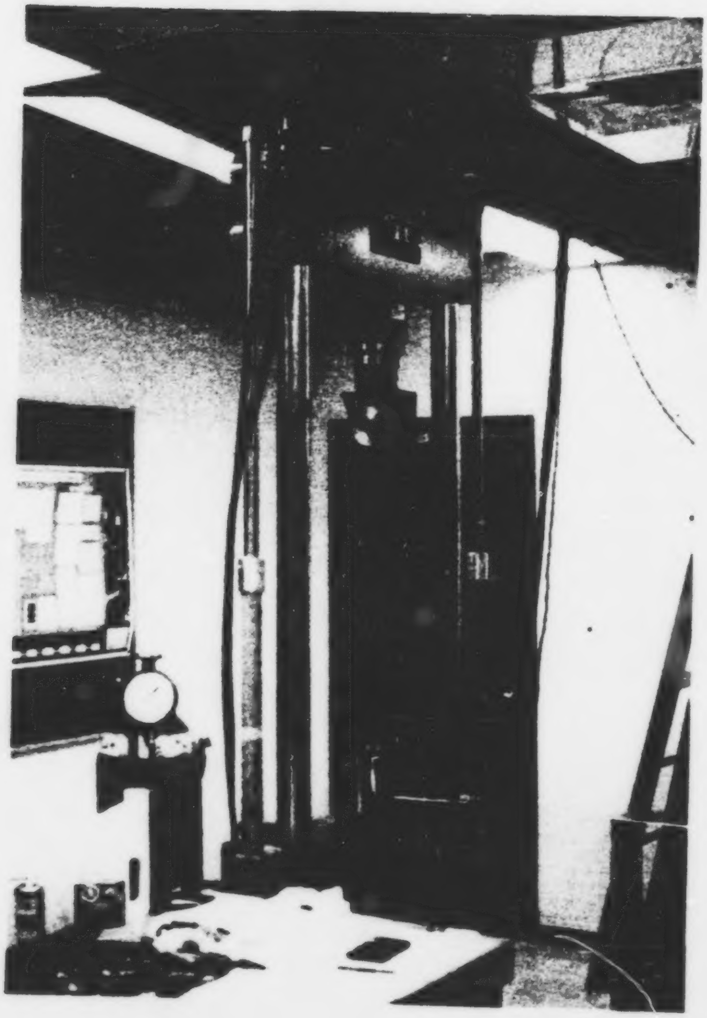


FIGURE 4.3: RADIUS ROD

radius rod mounting nuts were torqued to approximately 200 ft. lb. This held the inner steel sleeve of the radius rod bushing rigidly in place.

With the radius rod securely in place, the zero load position was found by observing the load cell output through a digital voltmeter.

The load on the rod was cycled sinusoidally about the zero load position ± 22 KN (± 5000 lb) at frequencies ranging from quasi-static to 1.5 Hz and at reduced amplitudes up to 4.0 Hz.

Results were recorded on the pen plotter on graphs of force versus displacement.

4.5.3 Air Spring

Two mounting plates were secured in the MTS machine with a number of bolts. The two ends of the air bag were secured to these plates and the mounting nuts were tightened. Air fittings were connected to the air bag, and approximately 6.5 feet of 1/4" O.D. plastic hose connected an air pressure gauge and an air valve stem to the air spring.

With the air bag set to some nominal height, pressurized air was added to the spring by the valve stem until a certain pressure was achieved. The nominal load was recorded from the load cell, and then the spring was cycled sinusoidally about the nominal position at several frequencies (0.01 - 5.0 Hz) and at several amplitudes ($\pm$ 4 mm - $\pm$ 20 mm). As well the air bag was tested at three nominal heights (standard ride height 273 mm; at 299 mm and at 248 mm). The data was recorded on a force versus displacement graph.

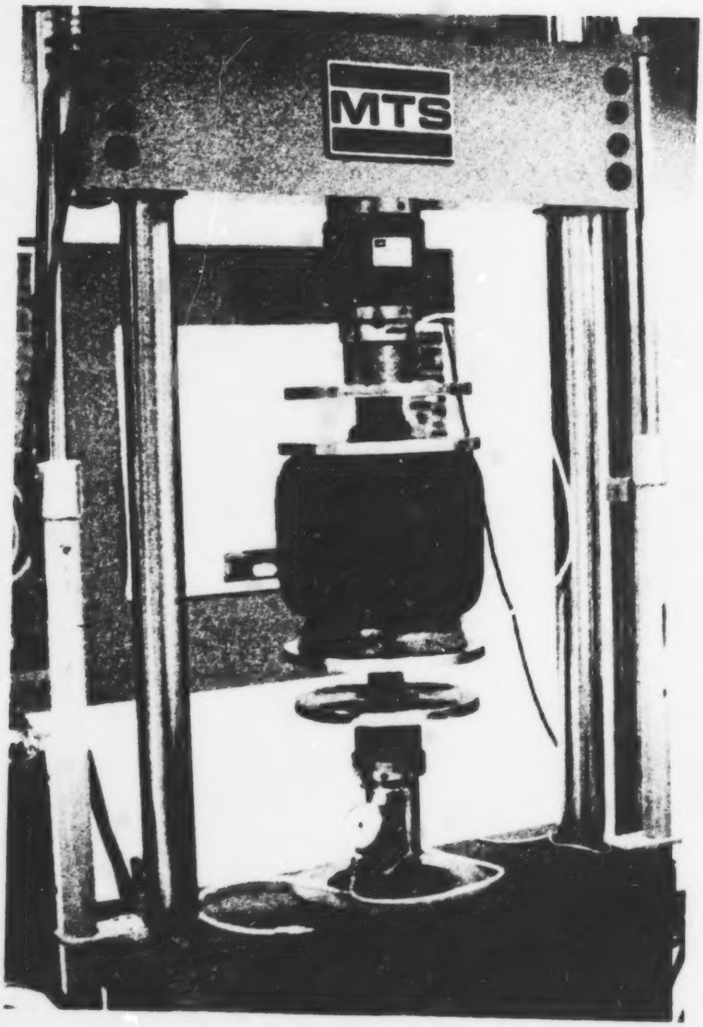


FIGURE 4.4: FRONT AIR SPRING

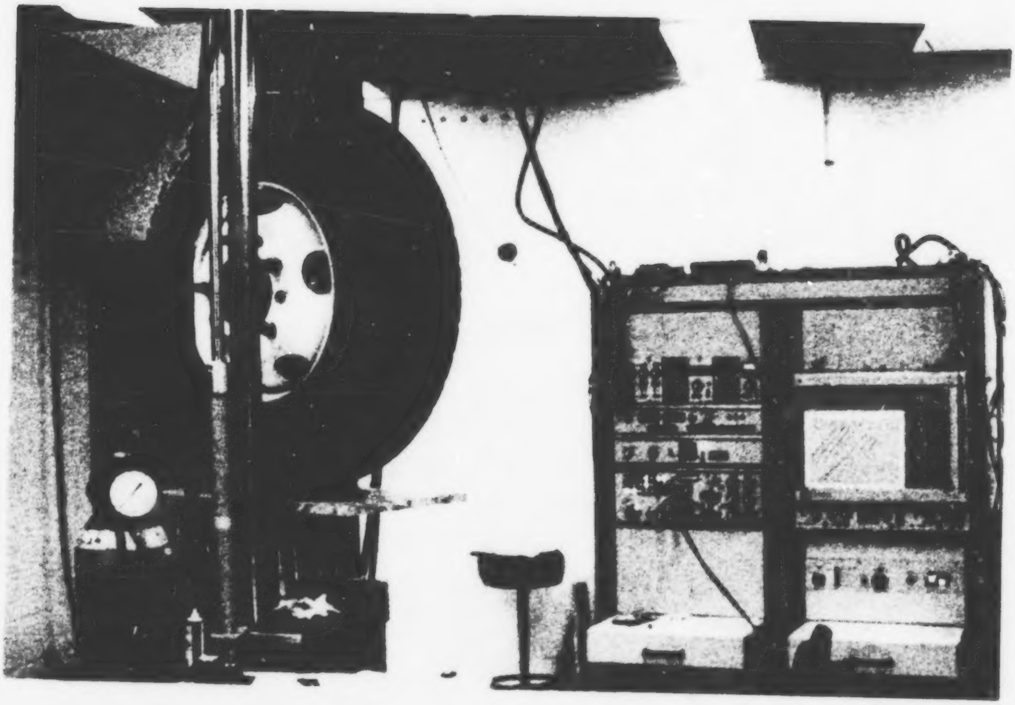


FIGURE 4.5: STANDARD BUS TIRE

Next, a 555 cu.in. steel reservoir was added at the end of the 6.5 ft. plastic hose. The above tests were repeated with the reservoir connected.

4.5.4 Tire

The tire mounting apparatus consisted of a welded steel structure with a 5 stud mounting plate for the wheel rim and an aluminum flat plate used as the loaded surface for the tire patch. The steel structure and flat plate were initially mounted in the MTS machine. Using a hoist,the tire was raised and positioned; after the wheel meshed with the 5 mounting studs, the nuts were torqued to approximately 300 ft. lb each.

With the testing rig assembled, the tire pressure was set to 100 psig for the unloaded tire. Slowly, the tire was compressed using the MTS controls until the maximum load of approximately 30 KN (6,744 lb) was reached and then the tire was unloaded again and the results of force versus displacement were recorded on the pen plotter. Next, the tire was loaded to an intermediate load of 20 KN ($\approx$ 5,000 lb) and the load was cycled sinusoidally plus or minus 10 KN to 1.0 KN about the nominal load at various frequencies. The frequencies varied from 0.1 Hz to 6 Hz. Finally, several different tire pressures were used (100, 90, 80, 70 psig).

Two different tires were tested:

1) A standard bus tire (Michelin XZA 12R 22.5) with approximately 3,000 km of use.

2) A low profile truck tire (Michelin XZA-1 275/80R) with no previous mileage.

4.6 METHOD OF ANALYSIS

4.6.1 Shock Absorbers

Figure 4.6 depicts a typical force versus displacement curve for a bus shock absorber cycled at different frequencies. Point A is the primary point of concern. At this point, the displacement is zero; the velocity is maximum; and the acceleration is also zero. Since the acceleration is zero, "inertia forces" are zero, and thus Point A establishes one point on a force/velocity graph of this damper. By taking the maximum cycle forces at the maximum cycle speeds for a number of different frequencies, force/velocity curves can be created. The above also applies to Point B where the damper is in compression.

4.6.2 Radius Rod

Force versus displacement curves are shown in Figure 4.7 for the front axle upper left radius rod. Stiffness measurments were made from these graphs by measuring the "average" slope of these curves (slope of line A-B). Curves in which the amplitude was decreased were analyzed, however, they should not be directly compared on a frequency basis to curves with different amplitudes.

4.6.3 Air Spring

The stiffness of the air spring was measured by the same technique as that of the radius rod. (Figure 4.8)

4.6.4 Tires

The stiffness of the tires were measured by calculating the slope of the force-displacement tire curves. (Figure 4.9)

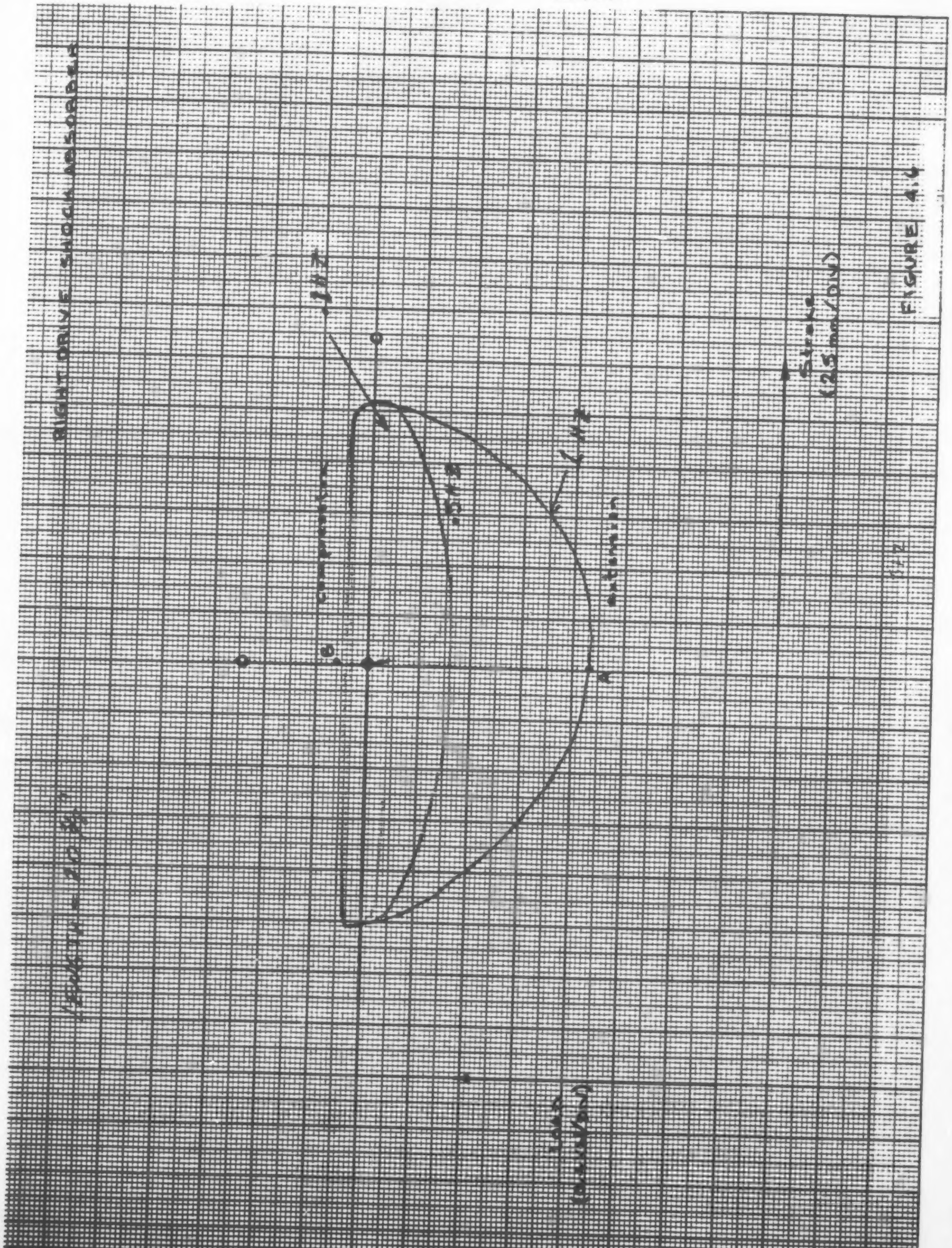

RIGHT DRIVE SHOCK ABSORBER
(LENGTH = 20.8")
.1 Hz
.5 Hz
1 Hz
compression
extension
B
A
Stroke (25 mm/DIV)
Load (2kN/DIV)

FIGURE 4.4

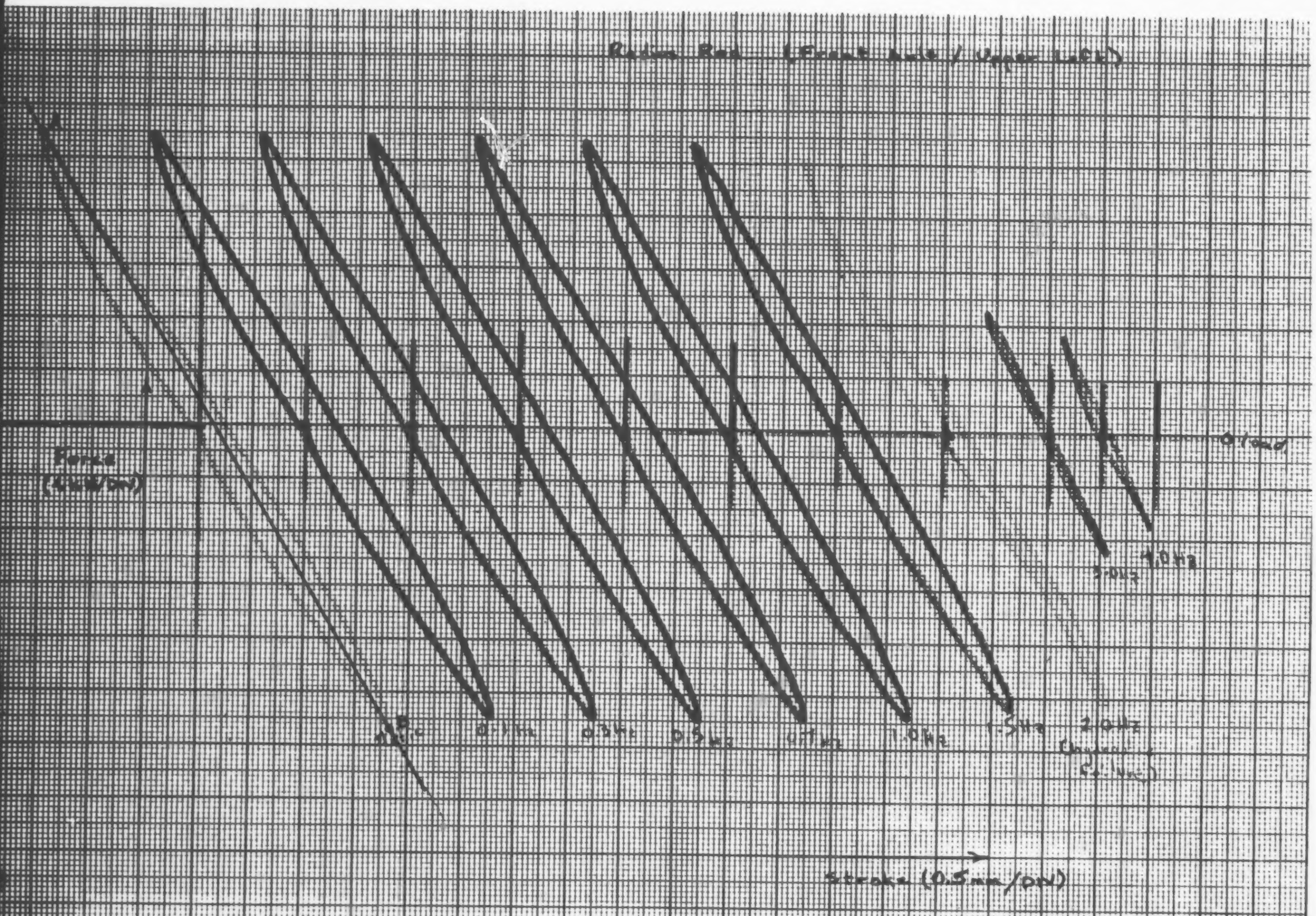

Radius Rod (Front Axle / Upper Left)
Force (4kN/DIV)
0 Load
0.1Hz
0.3Hz
0.5Hz
0.7Hz
1.0Hz
1.5Hz
2.0Hz
3.0Hz
4.0Hz
Stroke (0.5mm/DIV)
RR

FIGURE 4.7

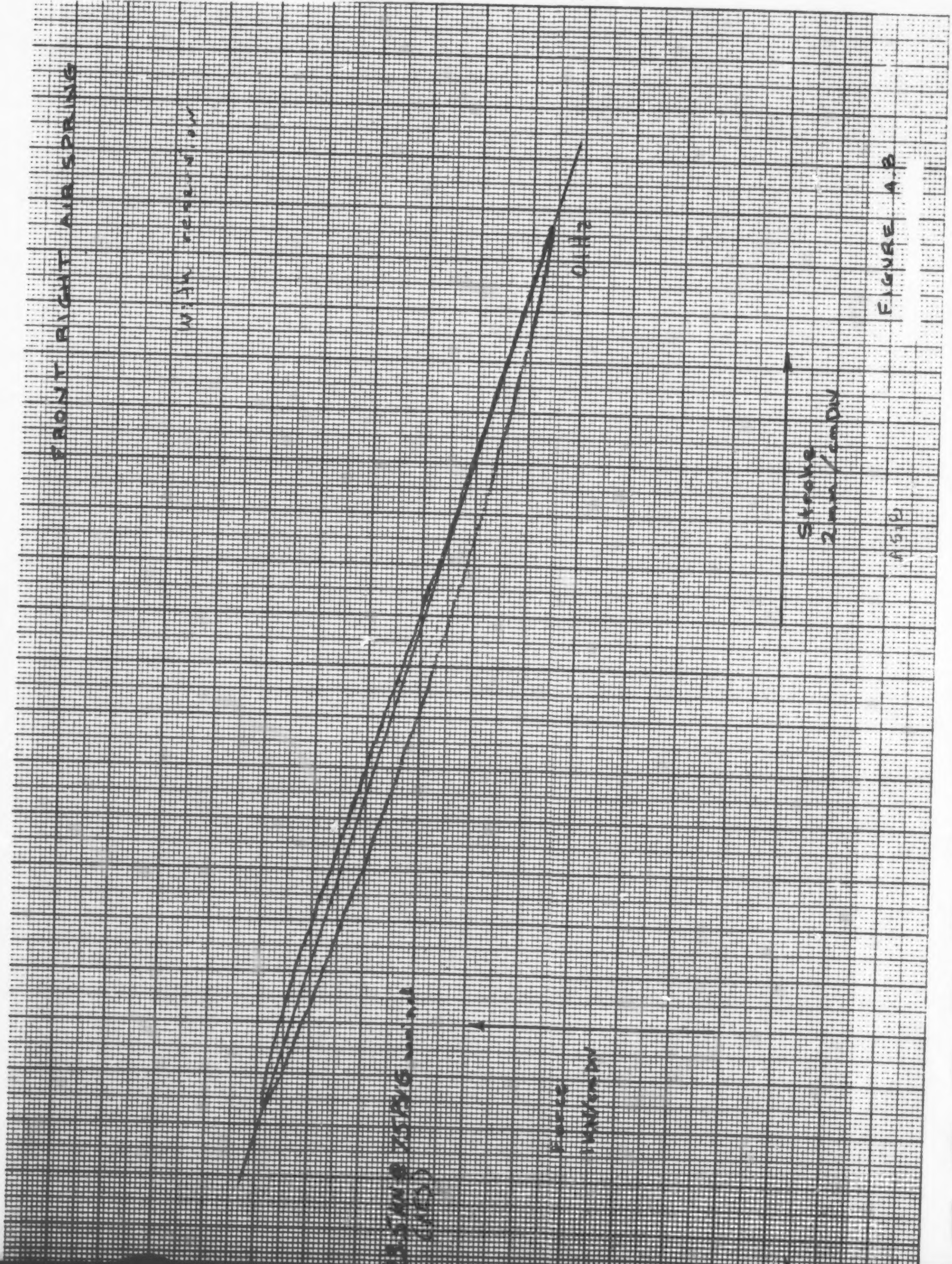
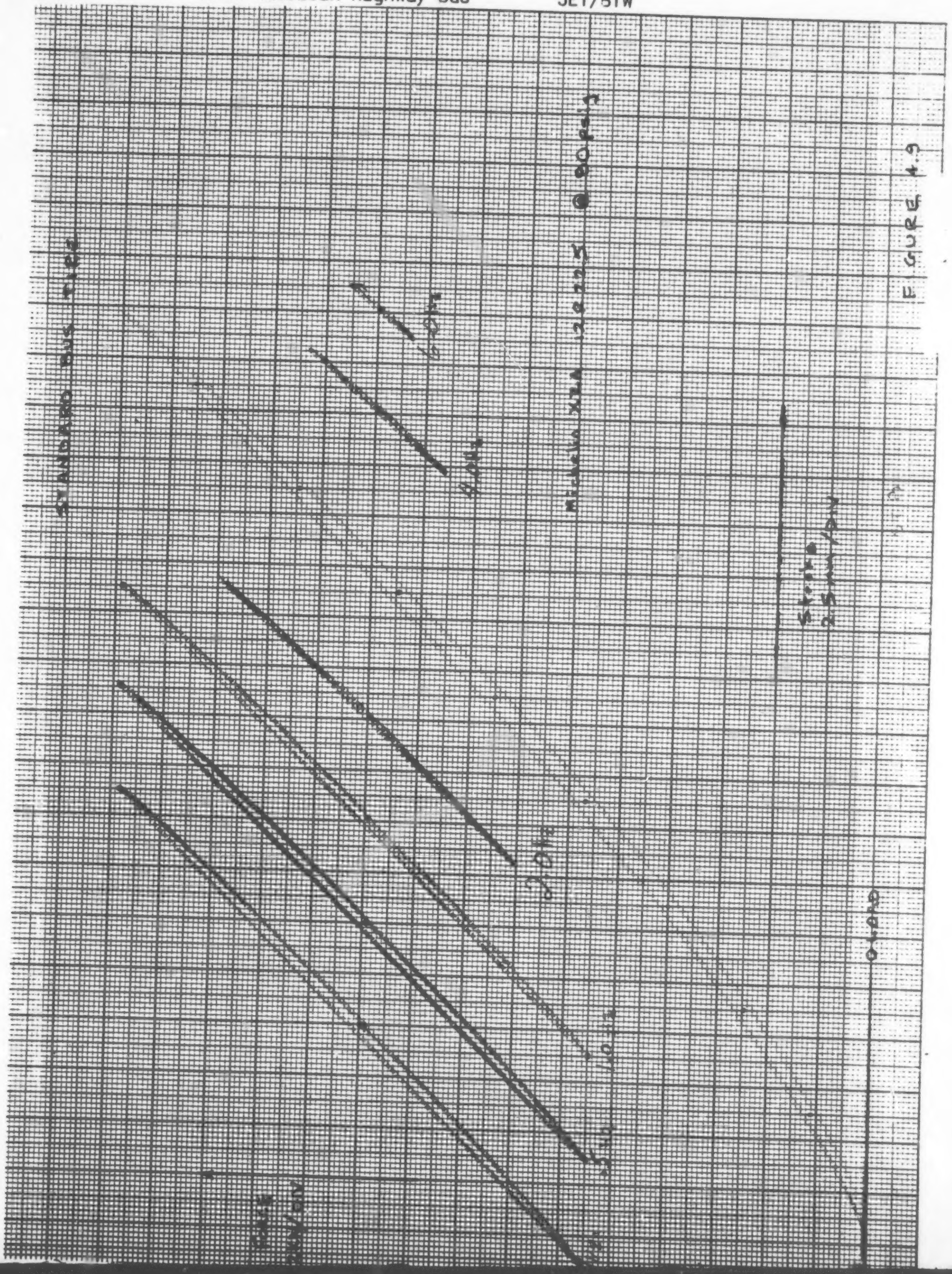

FRONT RIGHT AIR SPRING
With reservoir
0.1 Hz
3.5 kN @ 75 PSIG nominal
Force
1 kN/cm DIV
Stroke
2 mm/cm DIV
A5.8

FIGURE A.8



STANDARD Bus TIRE
Michelin X2A 12R22.5 @ 80 psi
FIGURE 4.9
6.0Hz
4.0Hz
2.0Hz
1.0Hz
Stroke 2.5mm/Div
0.400

4.7 RESULTS

The results are presented in Figures 4.10 - 4.18

4.7.1 Shock Absorber (See Figure 4.10)

The shock absorber force was found to be primarily a function of relative velocity between the two ends of the shock and shock temperature(1) for any shock absorber. Also, each shock absorber behaved very differently in compression than extension. In compression, forces through the shock were quite small and tended to vary linearly with velocity; as well the temperature effects in compression did not appear too significant. In extension, the forces were much larger and were more non-linear with velocity (usually a 2nd order polynomial fit worked quite well). Temperature effects in extension were quite significant ($\approx$ 30% change in force for a temperature change from 12°C to 28°C); however, the changes occurred most rapidly in the temperature range of 12°C to 16°C. Coloumb friction in all dampers was very similar and was found to be approximately 40 lb in both extension and compression.

(1) "Shock temperature" should not be taken as an absolute value, since it merely gives an indication of the casing temperature, not the actual oil temperature within the shock. It's purpose is primarily to show temperature changes.

4.7.2 Radius Rod (See Figure 4.11)

The radius rod data produced a static stiffness for the rubber bushings of 77,000 lb/in. The value quoted by the manufacturer was 75,000 lb/in. A dynamic spring rate was also quoted by the manufacturer of 1.5 times the static rate. Unfortunately, the MTS machine was not capable of achieving these results; however, a slight increase of approximately 5% the static stiffness was noted for higher frequencies.

4.7.3 Air Spring

Without a reservoir connected, the air bag stiffness was found to be 1,130 lb/in and constant over the frequency range 0 - 5 Hz for the conditions given in Figure 4.12. With the reservoir connected, the stiffness varied from 750 lb/in statically to 1,240 lb/in for high frequencies.

4.7.4 Tire Data

From Figures 4.16 and 4.17, the compression of a tire from zero load to maximum load is non-linear (again a second order polyomial fits the data well). However, over shorter portions of the curves (±1/4") a constant stiffness could be assumed.

The frequency of oscillation had some effect on the stiffness of the tires as is seen in Figure 4.18. As frequency increased, the stiffness increased. It appears that the standard bus tires stiffness was more affected by oscillations than the low profile truck tire.

The vertical stiffness of the tires at 100 psi tire pressure were the following:

Low profile tires:	5,360 lb/in
Standard tires:	5,330 lb/in

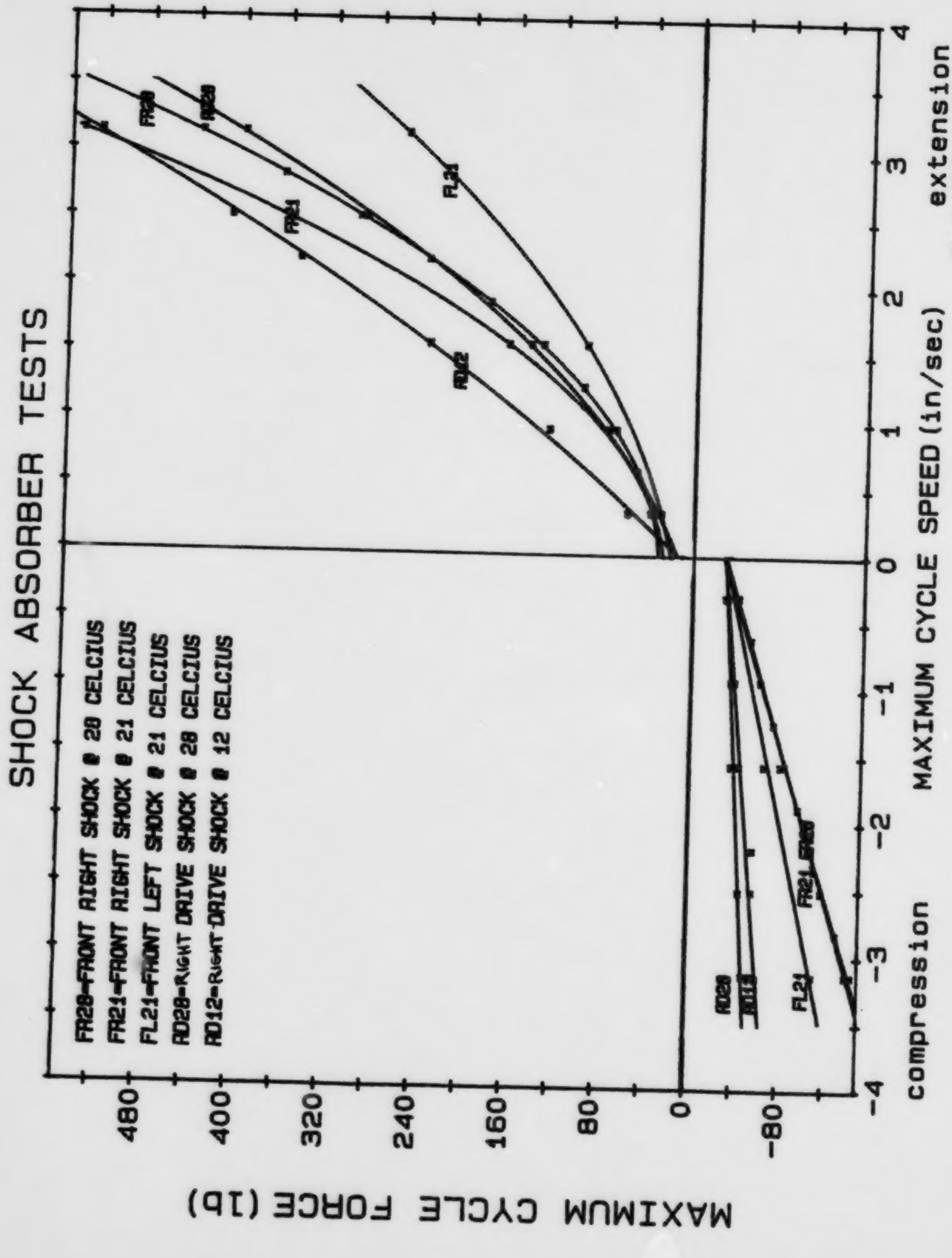

SHOCK ABSORBER TESTS
FR28-FRONT RIGHT SHOCK @ 28 CELCIUS
FR21-FRONT RIGHT SHOCK @ 21 CELCIUS
FL21-FRONT LEFT SHOCK @ 21 CELCIUS
RD28-RIGHT DRIVE SHOCK @ 28 CELCIUS
RD12-RIGHT DRIVE SHOCK @ 12 CELCIUS
MAXIMUM CYCLE FORCE (lb)
480
400
320
240
160
80
0
-80
MAXIMUM CYCLE SPEED (in/sec)
-4
-3
-2
-1
0
1
2
3
4
compression
extension
FR28
RD28
FR21
FL21
RD12
FR21 FR28
RD28
RD12
FL21

FIGURE 4.10

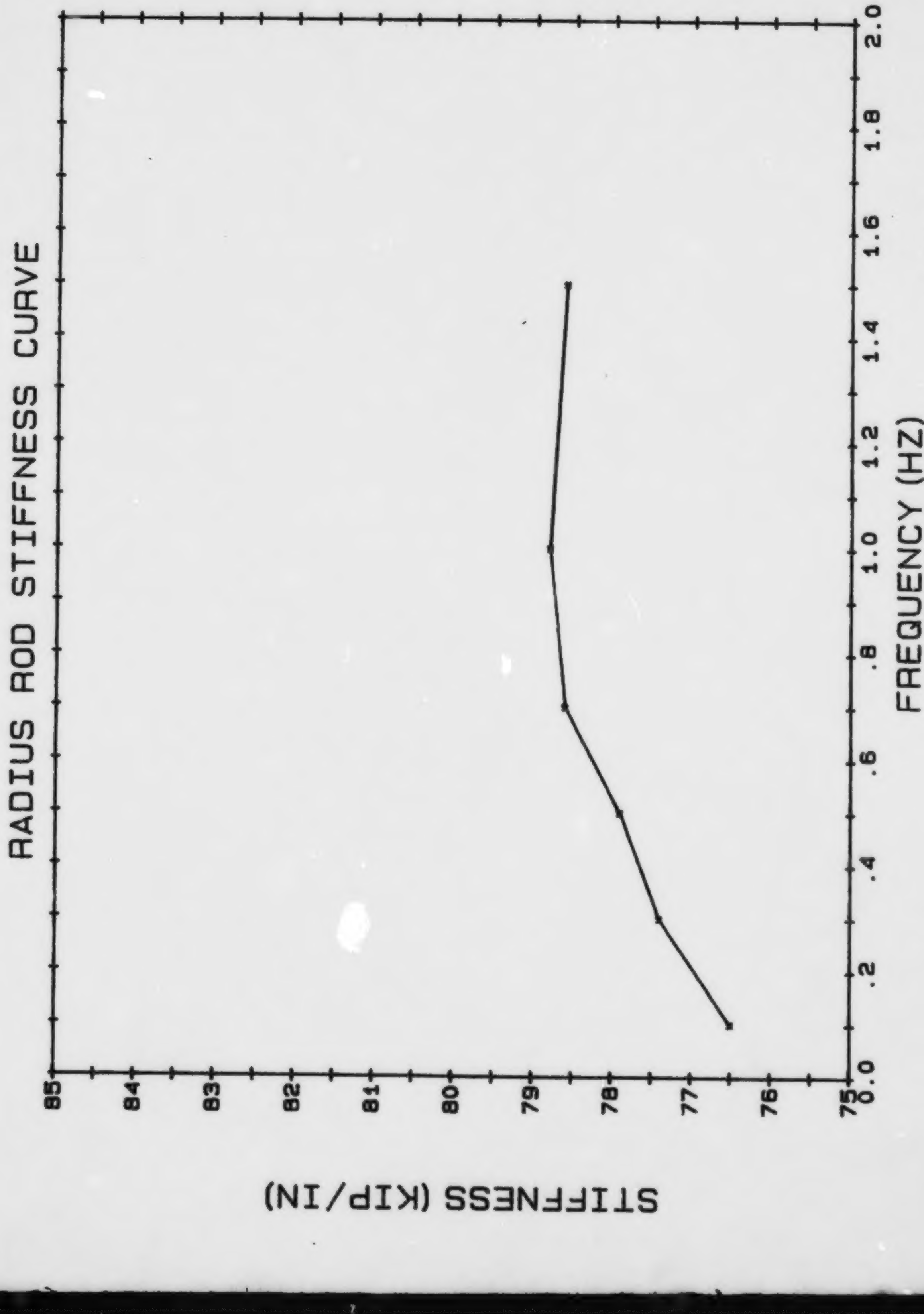

RADIUS ROD STIFFNESS CURVE
STIFFNESS (KIP/IN)
FREQUENCY (HZ)
85
84
83
82
81
80
79
78
77
76
75
0.0
.2
.4
.6
.8
1.0
1.2
1.4
1.6
1.8
2.0

FIGURE 4.11

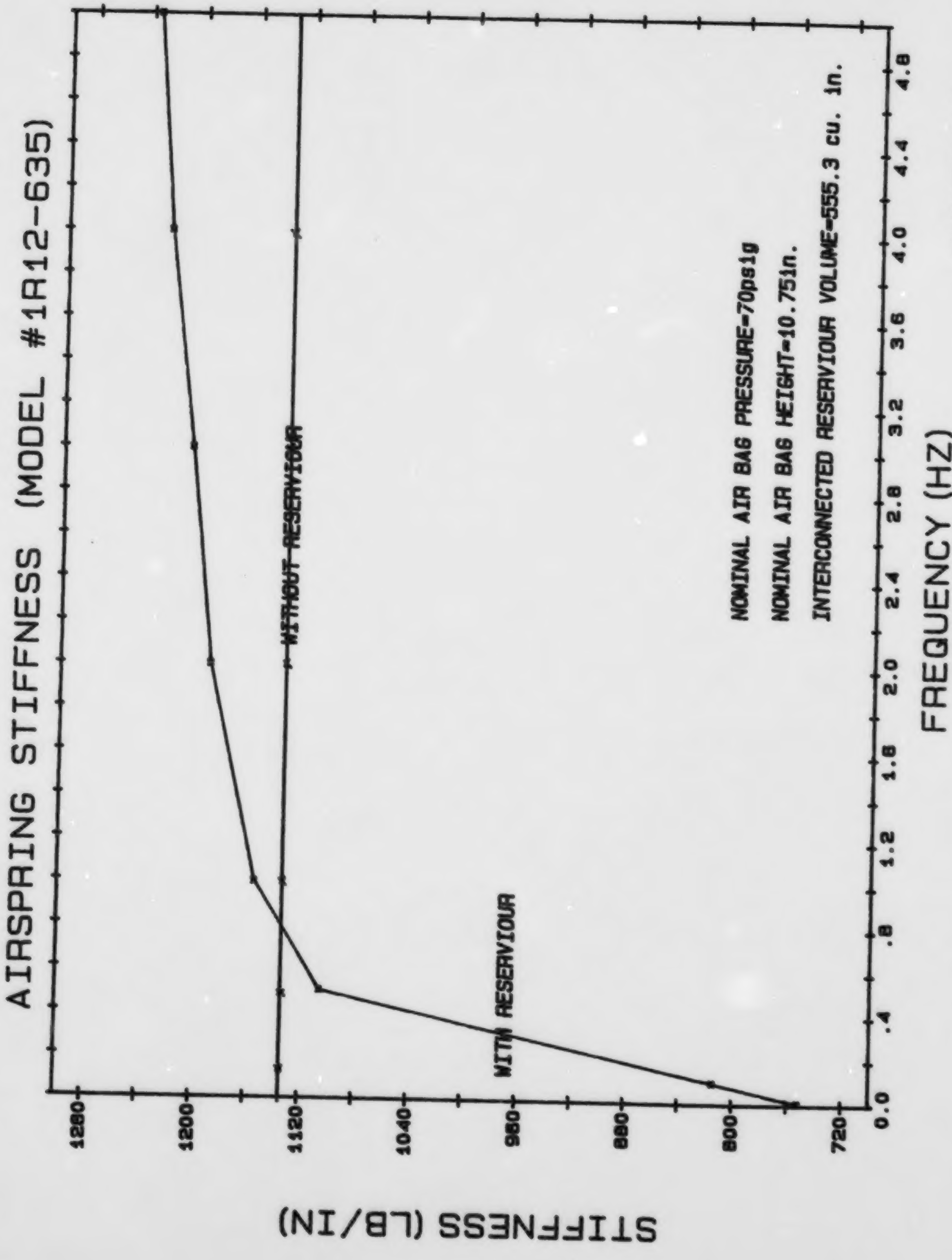

AIRSPRING STIFFNESS (MODEL #1R12-635)
STIFFNESS (LB/IN)
1280
1200
1120
1040
960
880
800
720
FREQUENCY (HZ)
0.0
.4
.8
1.2
1.6
2.0
2.4
2.8
3.2
3.6
4.0
4.4
4.8
WITHOUT RESERVIOUR
WITH RESERVIOUR
NOMINAL AIR BAG PRESSURE=70psig
NOMINAL AIR BAG HEIGHT=10.75in.
INTERCONNECTED RESERVIOUR VOLUME=555.3 cu. in.

FIGURE 4.12

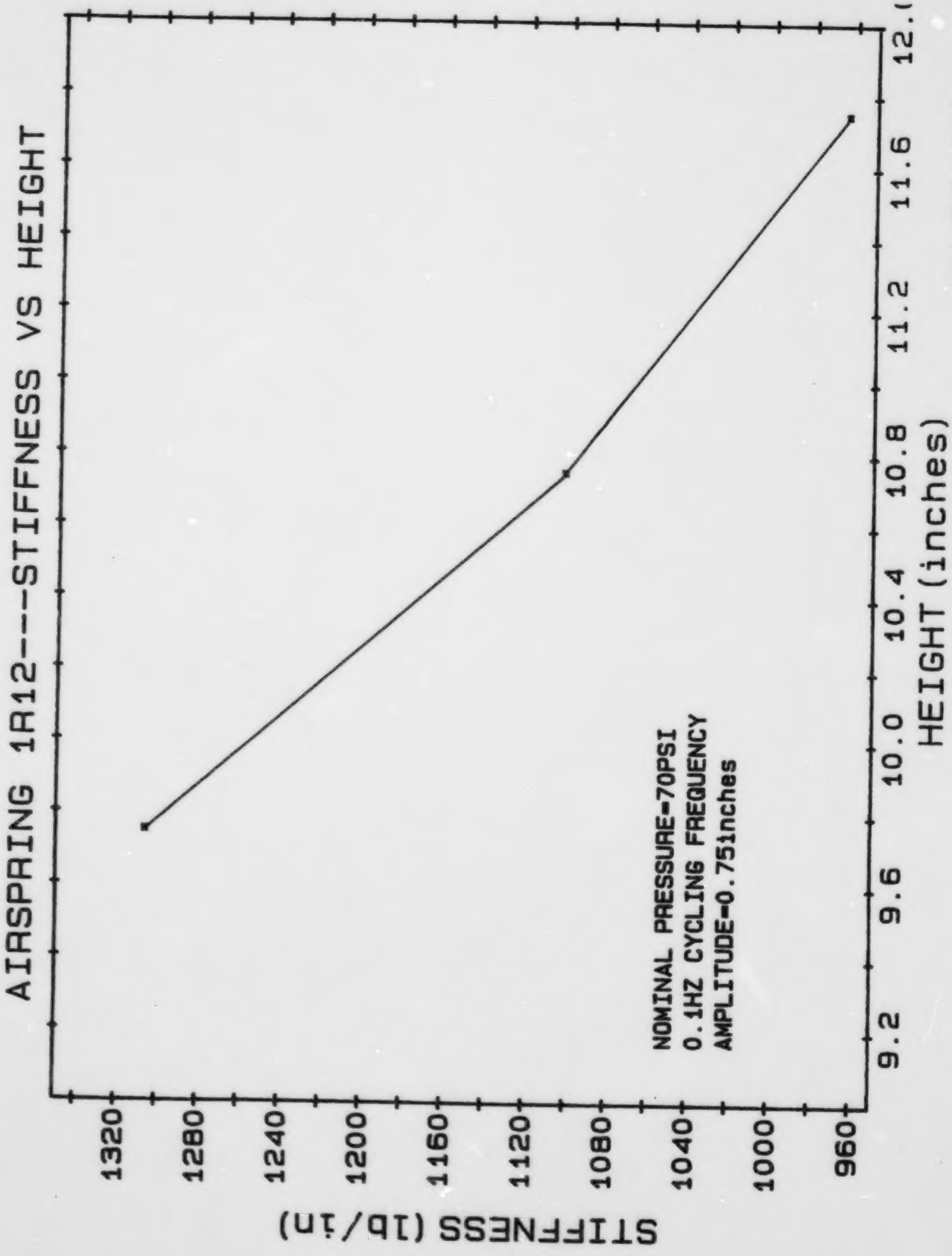

AIRSPRING 1R12---STIFFNESS VS HEIGHT
STIFFNESS (lb/in)
1320
1280
1240
1200
1160
1120
1080
1040
1000
960
HEIGHT (inches)
9.2
9.6
10.0
10.4
10.8
11.2
11.6
12.(
NOMINAL PRESSURE=70PSI
0.1HZ CYCLING FREQUENCY
AMPLITUDE=0.751inches

FIGURE 4.13

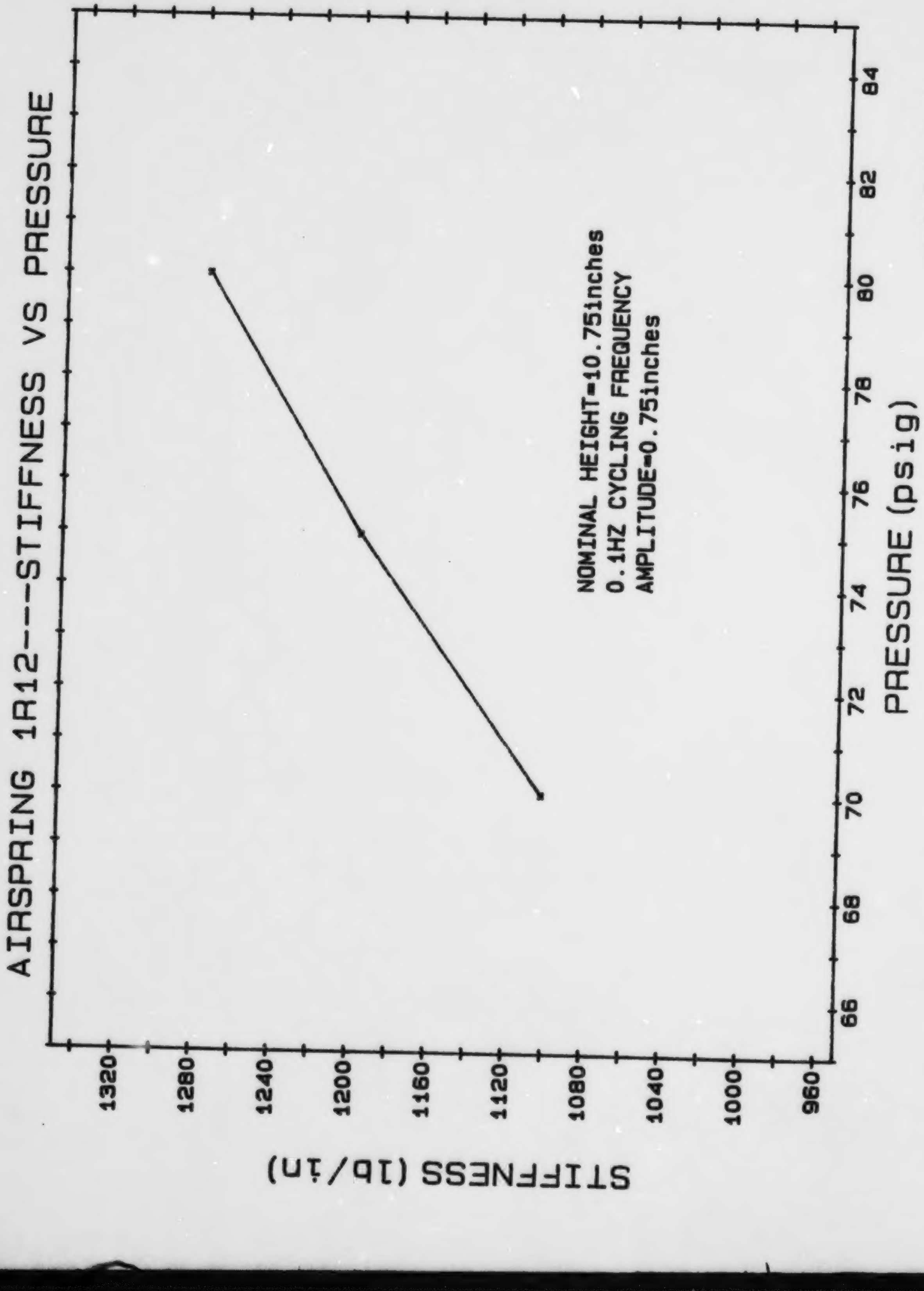

AIRSPRING 1R12---STIFFNESS VS PRESSURE
NOMINAL HEIGHT=10.75inches
0.1HZ CYCLING FREQUENCY
AMPLITUDE=0.75inches
PRESSURE (psig)
66
68
70
72
74
76
78
80
82
84
STIFFNESS (lb/in)
960
1000
1040
1080
1120
1160
1200
1240
1280
1320

FIGURE 4.14

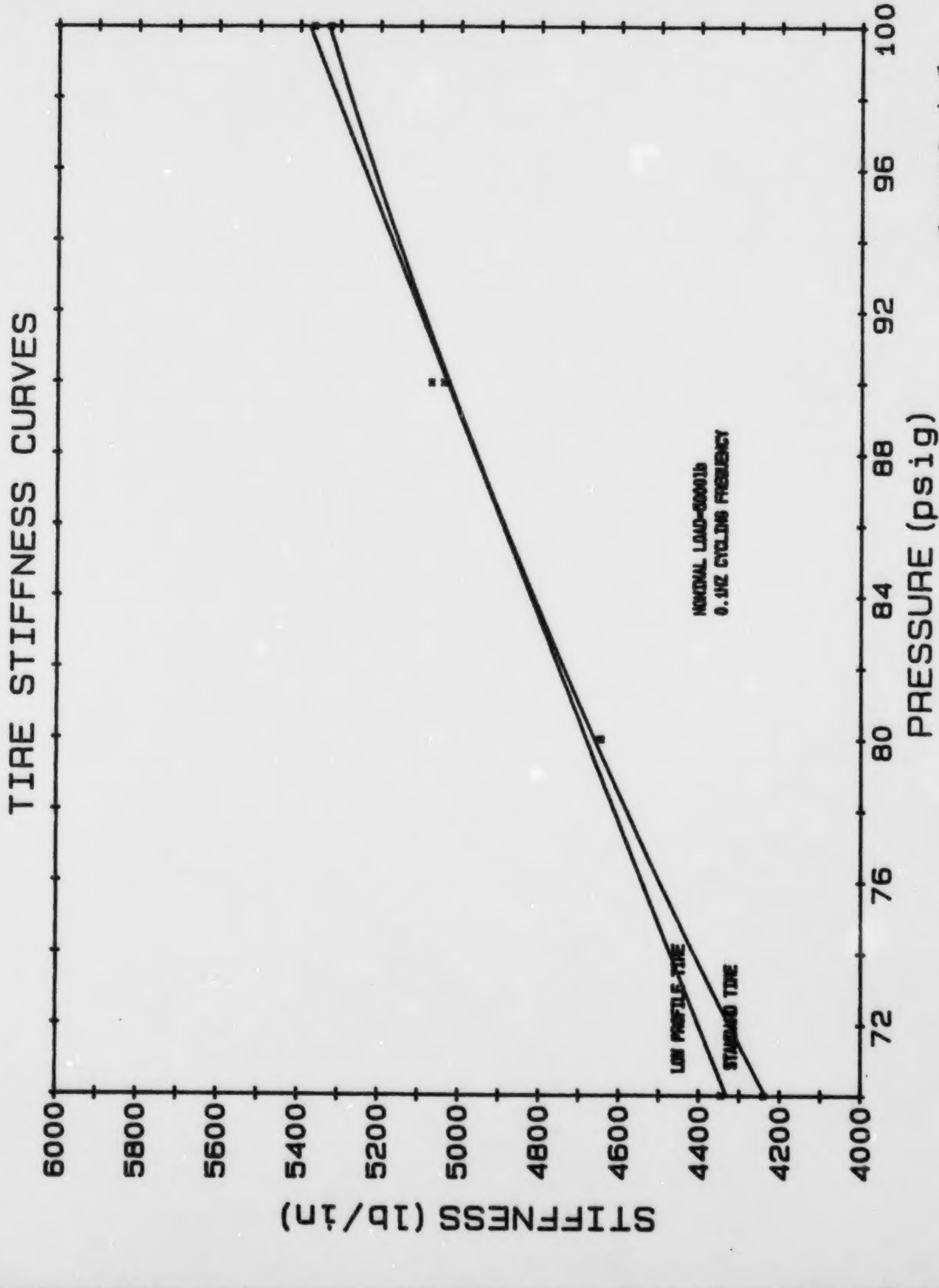

TIRE STIFFNESS CURVES
NOMINAL LOAD-5000lb
0.1HZ CYCLING FREQUENCY
LOW PROFILE TIRE
STANDARD TIRE
STIFFNESS (lb/in)
6000
5800
5600
5400
5200
5000
4800
4600
4400
4200
4000
PRESSURE (psig)
72
76
80
84
88
92
96
100

FIGURE 4.15

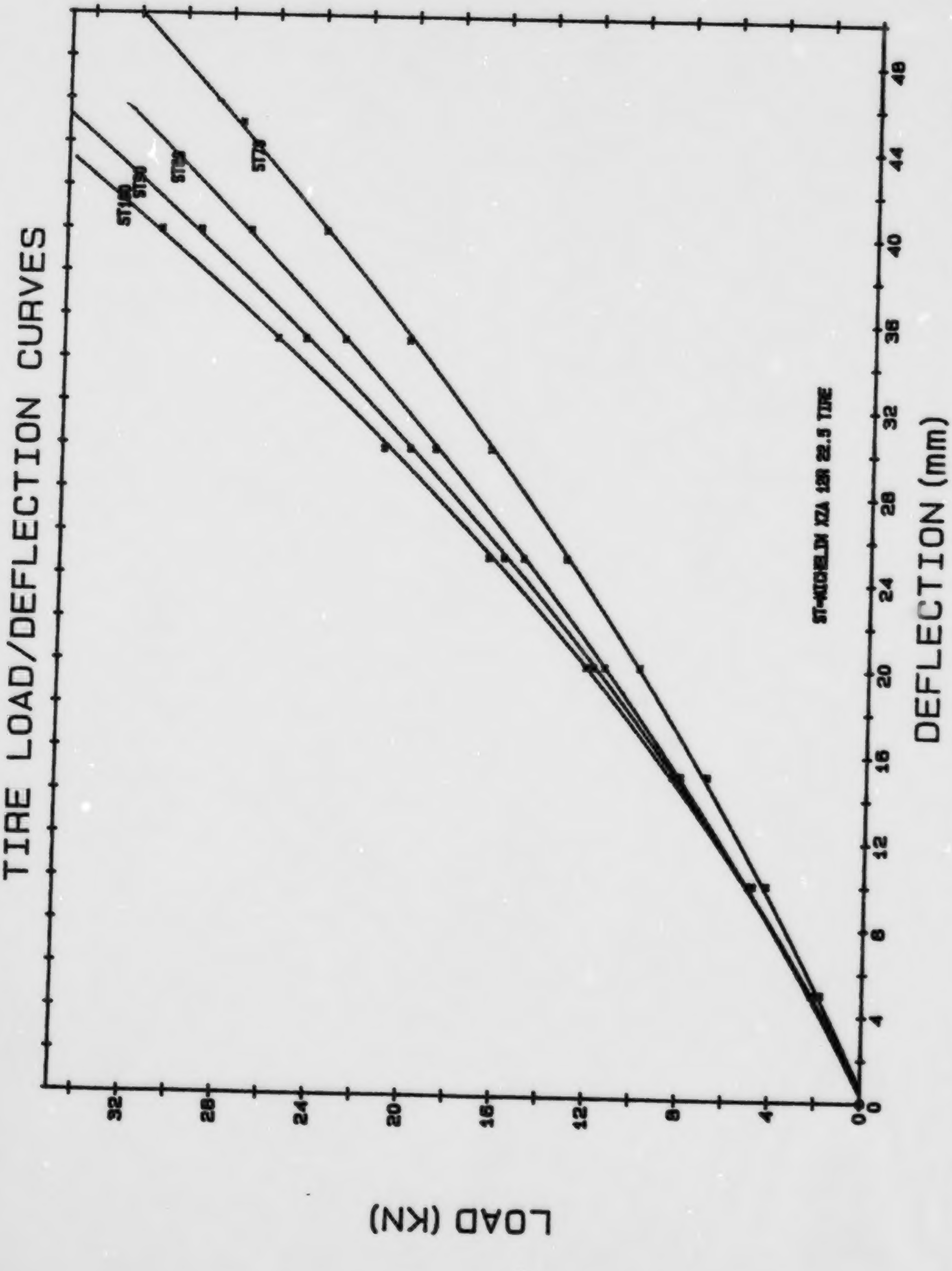

TIRE LOAD/DEFLECTION CURVES
LOAD (KN)
0
4
8
12
16
20
24
28
32
DEFLECTION (mm)
0
4
8
12
16
20
24
28
32
36
40
44
48
ST-MICHELIN XZA 12R 22.5 TIRE
ST70
ST80
ST90
ST100

FIGURE 4.16

TIRE LOAD/DEFLECTION CURVES

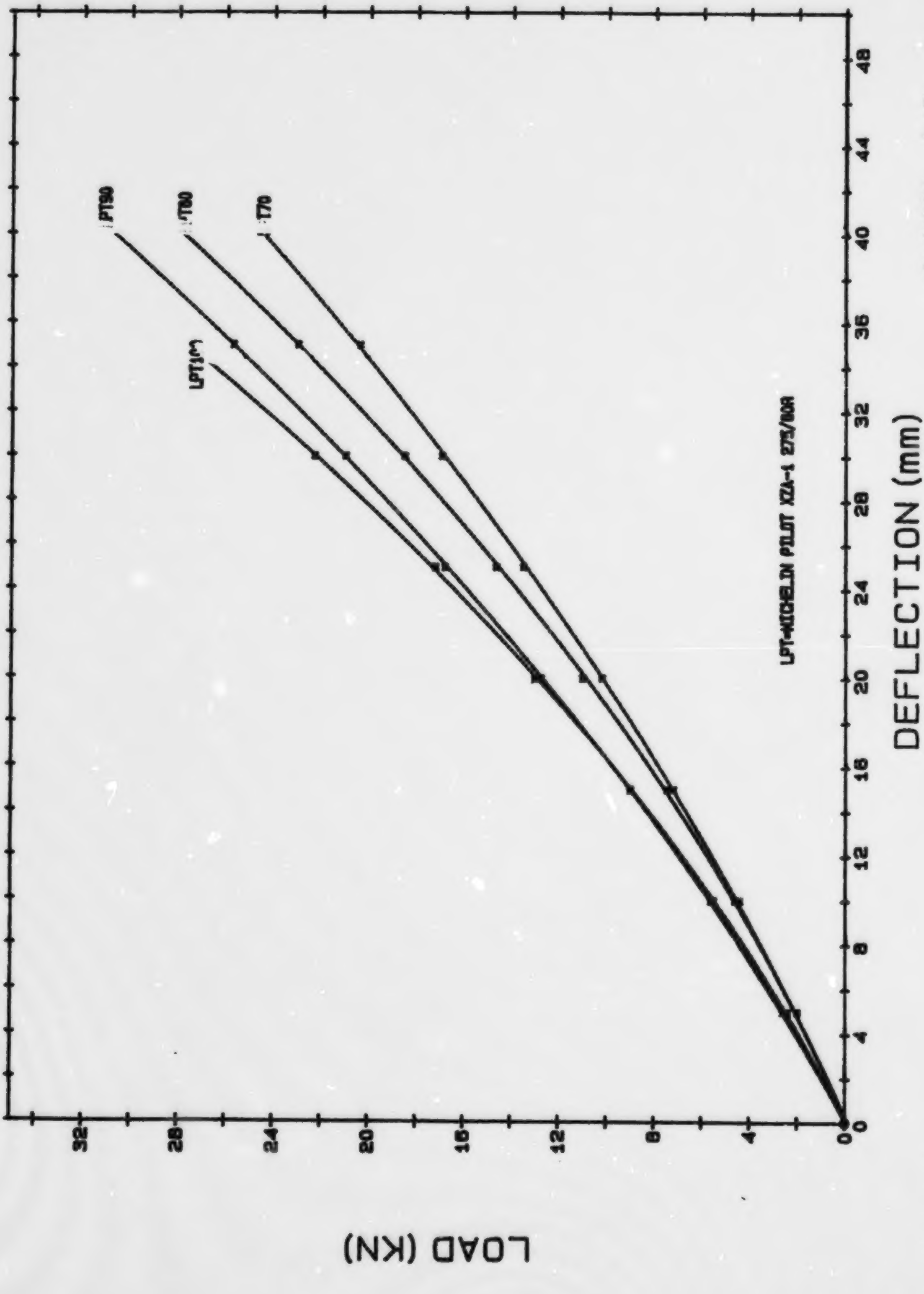


FIGURE 4.17

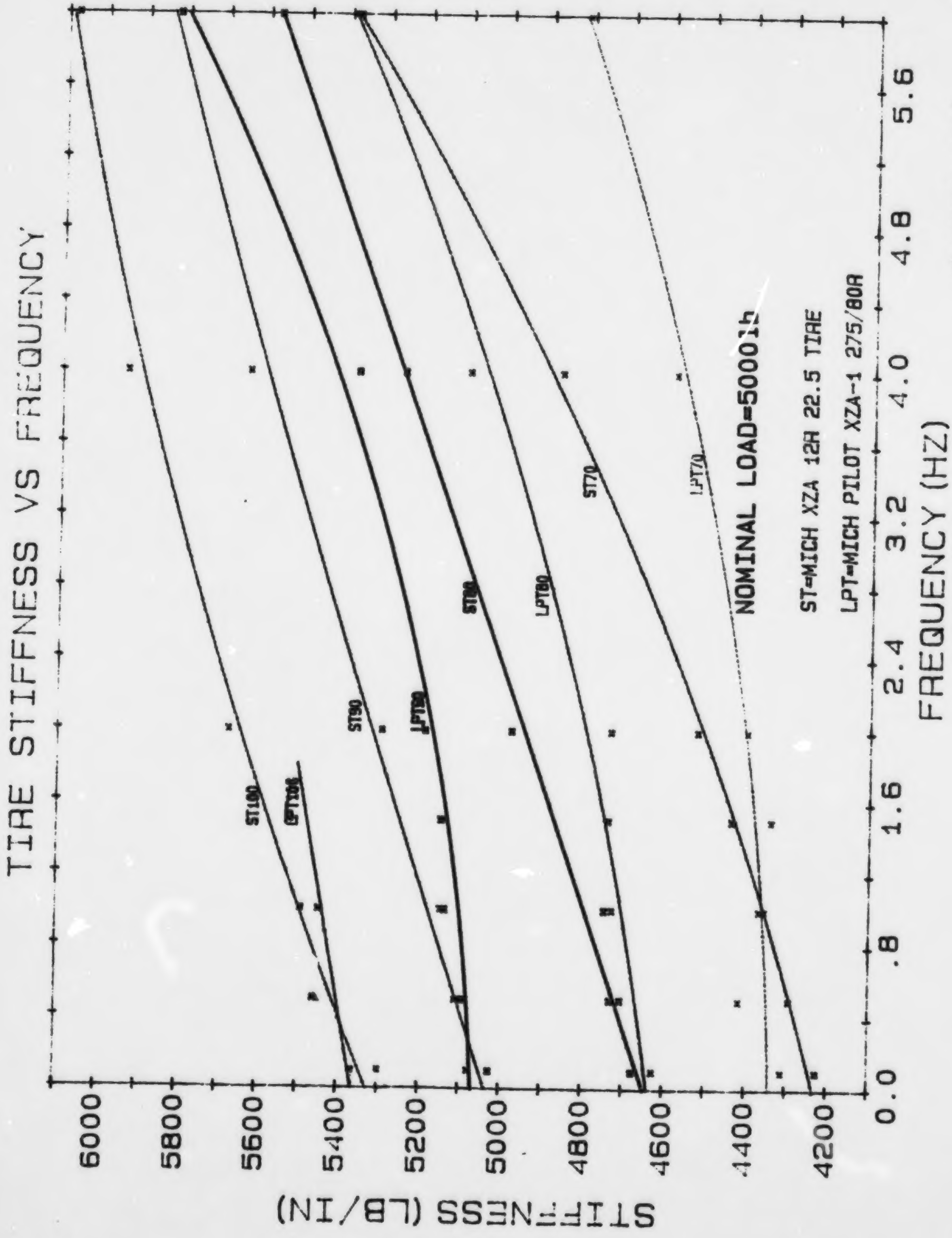

TIRE STIFFNESS VS FREQUENCY
STIFFNESS (LB/IN)
FREQUENCY (HZ)
6000
5800
5600
5400
5200
5000
4800
4600
4400
4200
0.0
.8
1.6
2.4
3.2
4.0
4.8
5.6
ST100
LPT100
ST90
LPT90
ST80
LPT80
ST70
LPT70
NOMINAL LOAD=5000lb
ST=MICH XZA 12R 22.5 TIRE
LPT=MICH PILOT XZA-1 275/80R

FIGURE 4.18

4.8 DISCUSSION AND CONCLUSIONS

4.8.1 Shock Absorbers

Using a second order curve fit routine, the following shock absorber characteristics were calculated:

where: $\text{Force (lb)} = A0 + A1\,(\text{velocity (in/sec)}) + A2\,(\text{velocity (in/sec)})^2$

	A0	A1	A2
COMPRESSION DATA			
Front Right (28°C)	-29.304	34.42	-0.2963
(21°C)	-32.446	24.558	-3.474
Front Left (21°C)	-32.694	16.747	-2.511
Right Drive (12°)	-22.391	19.579	2.646
(28°)	-30.178	3.232	-0.9518
EXTENSION DATA			
Front Right (28°C)	31.389	4.8185	39.742
(21°C)	27.415	12.176	47.997
Front Left (21°C)	20.731	21.644	16.963
Right Drive (12°)	14.315	117.58	14.703
(28°)	14.912	45.176	24.809

TABLE 4.1: SHOCK ABSORBER CURVE FITS

A very strong dependence was apparent between shock casing temperature and shock force. This result is quite reasonable when the change in viscosity of oil with temperature is considered. From the shock absorber results one could conclude that temperature has a very major effect on damping properties of shock absorbers.

Another significant result from this data is the large difference in damping properties between the right and left front shock absorbers. Considering that these dampers have the same model number, then it is apparent that one of the dampers is not functioning to its full potential.

4.8.2 Radius Rod

The longitudinal stiffness of the upper left front radius rod as measured agreed very closely (within 3%) with the manufacturers quoted stiffness of 75,000 lb/in.

4.8.3 Air Spring

Air spring stiffnesses were measured for various air spring conditions of height and pressure. The most interesting result from the air spring tests is the effect that the reservoir has. At very low frequencies, the air spring stiffness with reservoir is much lower then the stiffness for the air spring alone. This result is expected since adding a reservoir increases the air volume and since stiffness is inversely proportional to air volume, than a large decrease in stiffness is expected.

However, at higher frequencies, the stiffness of the air spring with reservoir increases. This can be rationalized when one considers that the air flow between the spring and the reservoir is not instantaneous and thus at higher frequencies the effective air volume which determines the spring stiffness is reduced. This increases the spring stiffness.

4.8.4 Tires

The tires were tested and stiffnesses were calculated for different pressures and cycling frequencies. Both types of tires (Michelin XZA 12R22.5 and Michelin Pilot XZA-1 275/80R) demonstrated increasing stiffness with increasing pressure: as well they both became stiffer with increasing cycling frequency.

One final point concerning the tire stiffnesses is that these stiffnesses are non-rolling. Rolling stiffnesses should be in the order of 5-10% lower.

5.0 RIDE QUALITY

The performance of the bus suspension system in providing good ride characteristics for the passengers was tested.

5.1 TEST OBJECTIVE

The objective was to measure the response of the test bus body (ride quality) to vertical inputs to the bus tires for a range of operating conditions. An additional objective was to measure axle response as complementary information for the validation of the computer simulation.

5.2 TEST APPROACH

Ride quality instrumentation was set-up to measure accelerations within the bus body, accelerations on the axles, displacements of the axles relative to the body and forward speed of the bus.

The vertical input at the bus tires was derived in three basic manners as given below.

5.2.1 Random Road Input

For Random Input, the bus was driven along preselected portions of highway (of varying roughness grades) at different speeds, while the ride instrumentation measured the relevant parameters.

5.2.2 Step Input

The bus was driven over a railway crossing which was sufficiently rough to simulate a 'step' and thus excite the suspension natural frequencies. The ride instrumentation measured necessary parameters as the test bus was driven at different speeds over the railway crossing.

5.2.3 Well - Defined Harmonic Input

The bus was placed on a four post shaker and the tag and drive wheels were excited at different frequencies, amplitudes and phases while the ride instrumentation recorded all meaningful parameters.

5.3 INSTRUMENTATION AND EQUIPMENT

Ride instrumentation consisted of the following transducers:

3 - ±2 g accelerometers Terra Technology Model #SA-102
2 - ±1 g accelerometers Statham
2 - ±5 g accelerometers Terra Technology Model #SA-102
2 - ±4 inch displacement transducer Collins corp. Model # D1603
2 - ±4 inch displacement transducer Inter Technology Model # SE373/100
1 - Speed transducer (operated off drive shaft) - built in-house

Additional Equipment

1 - AC inverter Topaz G7500
2 - DC power supplies Hp 8326B, Lambda
1 - Transducer power supply and signal attenuation junction box (built in-house)
1 - 14 channel Sangamo Magnetic Tape Instrumentation

Recorder/Reproducer with voice side channel Sabre VI

1 - Fluke digital multi-meter 8000A

- Appropriate lengths of four conductor cables
- MTS four post shaker (Capacity: 35 kip/post)

5.4 TEST SET-UP AND LOCATION

Five accelerometers were mounted in the bus body. The following describes their locations:

1. ±2 g accelerometer - vertical axis and close to c.g.,at floor level
2. ±2 g accelerometer - vertical axis and at seat post of right front seat; at floor level
3. ±2 g accelerometer- vertical axis and at seat post of left rear seat; at floor level.
4. ±2 g accelerometer - lateral axis and close to c.g., at head height
5. ±2 g accelerometer - longitudinal axis and close to c.g., at head height.

Two accelerometers were mounted on the test bus axles:

1. ±5 g accelerometer - vertical axis and mounted at the centre of the front axle.
2. ±5 g accelerometer - vertical axis and mounted on top of the differential of the drive axle.

Four displacement transducers were mounted to the test bus axles.

1. ±4 inch displacement transducer - measured right front vertical displacement of the axle relative to the bus body. One end was attached to an aluminum bracket on the bus body and the other end was attached to an aluminum bracket connected to the axle.

2. ±4 inch displacement transducer - same as right front transducer, except mounted to the left side of the front axle.

3. ±4 inch displacement transducer - measured right drive vertical displacement of the axle relative to the bus body. Brackets were attached to the top and bottom shock absorber mounting pins and the transducer was connected between these brackets.

4. ±4 inch displacement transducer - same as right drive displacement tranducer, except mounted to the left side of the axle.

The locations of these transducers in a global co-ordinate system are given in Appendix 2.

Ride Quality tests took place at several different locations. The following is a summary of these events.

June 6/86	Step input, Celanese Rail Crossing, County Rd #23, Configuration #3 (See Section 2.0).
June 9/86	Step input, Celanese Rail Crossing, County Rd #23, Configuration # 3

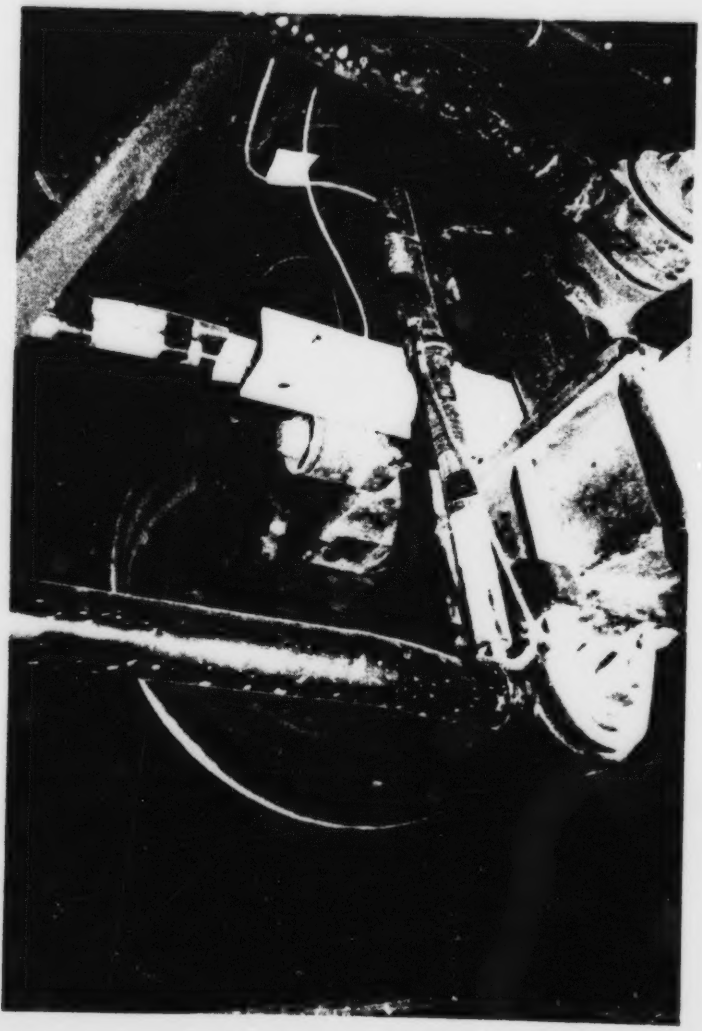


FIGURE 5.1: RIGHT FRONT AXLE DISPLACEMENT TRANSDUCER

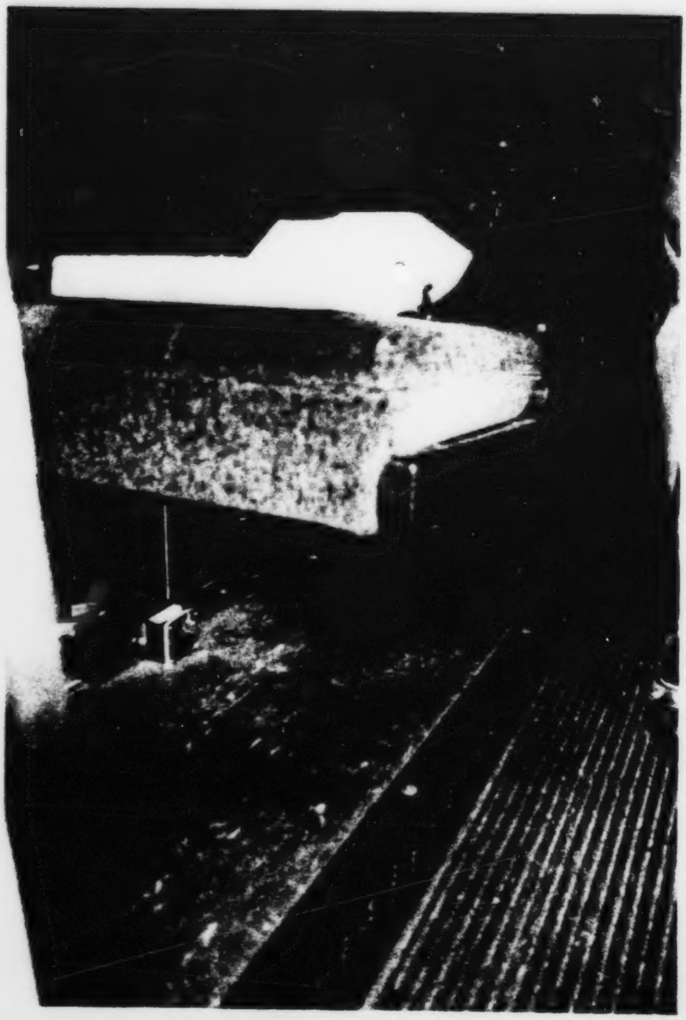


FIGURE 5.2: RIGHT FRONT SEAT ACCELEROMETER

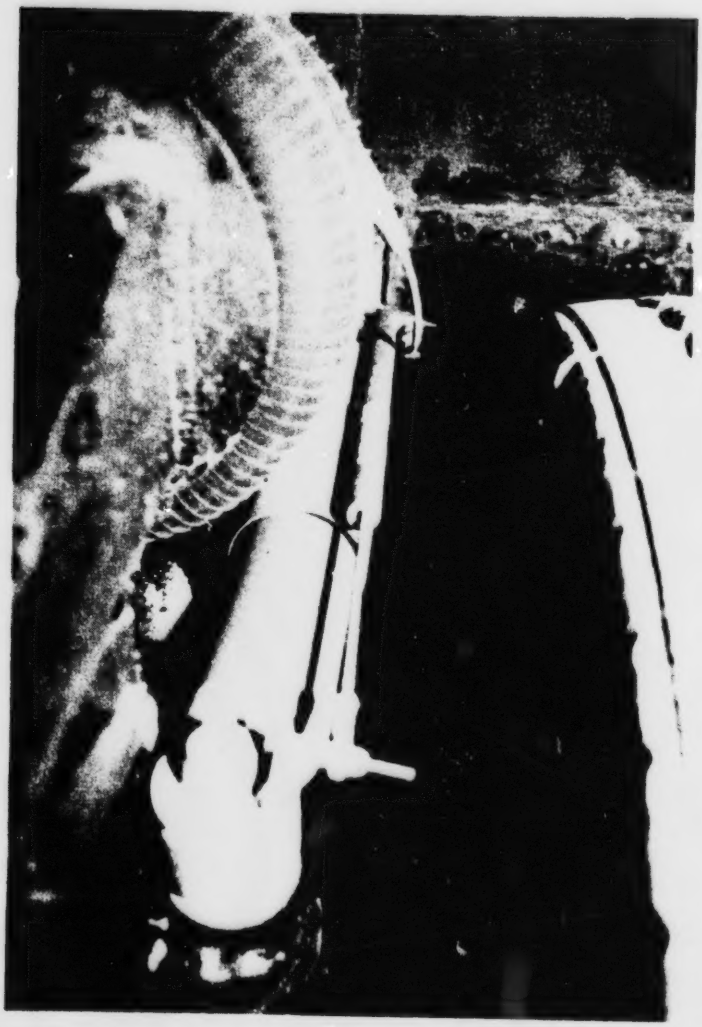


FIGURE 5.3: DRIVE AXLE DISPLACEMENT TRANSDUCER

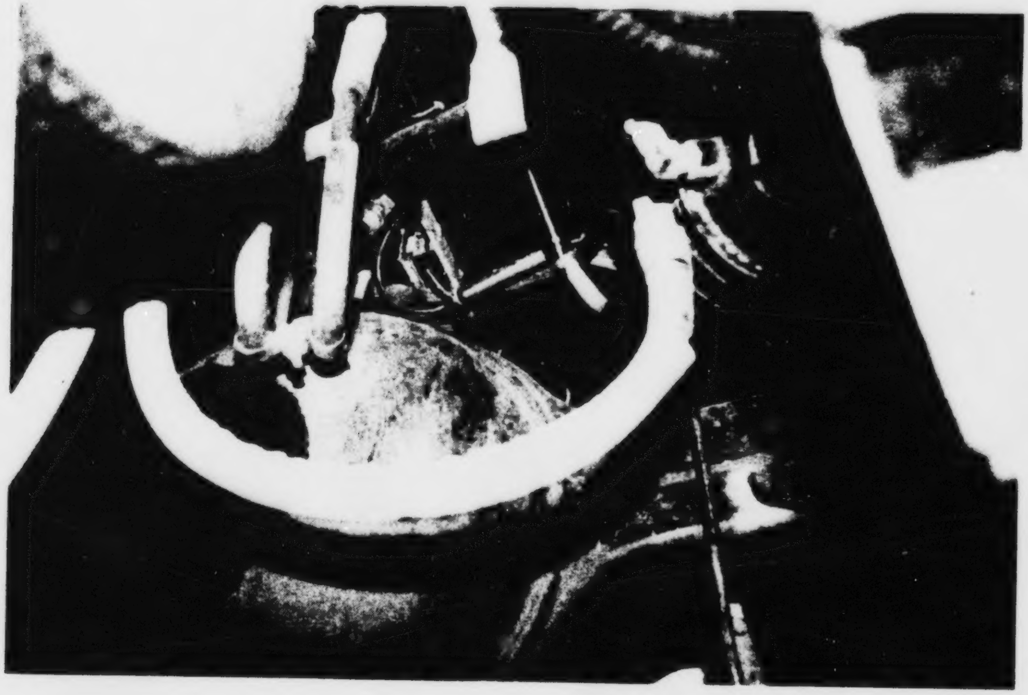


FIGURE 5.4: DRIVE AXLE ACCELEROMETER

June 10/86	Random Input, Highways #401 and 133, (en route to Bellevile, Ontario), Configuration #3.
June 18/86	Step input, Celanese Rail Crossing, County Rd #23, Random input, Highway #133, Configuration #3
June 19/86	Random input, Highways #401 and 133, (en route to Mountain View Airport), Configuration #3.
June 25/86	Well-defined input, NRC shaker, Railway Laboratory, Ottawa, Configuration #5.
June 26/86	Well-defined input, NRC shaker, Railway Laboratory, Ottawa, Configuration #5, 6, and 7.
July 2/86	Random input, Highway #133, Configuration #5.
July 7/86	Random input, Highway #2, #401 (en route to Blainville, Quebec), Configuration #8.
July 10/86	Random input, Highway #133, Configuration #10.
July 11/86	Random input, County Rd. #5, Configuration #11.
July 14/86	Random input, Highway #133, #401, County Rd. #4 and 5, Configuration #12, 13, and 15.
July 16/86	Random input, Highway #133, 401 and County Rd. #4 and 5, Configuration #16.

5.5 TEST PROCEDURES

The procedures for performing a Ride Quality test were the same regardless of the input (Random, Step or Well-defined). In each case all power supplies were turned on and all necessary connections to the junction box were checked to confirm the correct instrumentation for Ride Quality. Next, the excitation voltages from the power supplies were checked to confirm the correct levels (one supply at ±10V and the other at ±15V). All tape channels were then calibrated by simultaneously recording standard DC signals (+3.77V, 0.0V and -3.72V) to all channels. This was done usually at the beginning and end of each ride quality test. Any addition relevant information during a test was recorded on the voice channel of the Sangamo recorder.

5.5.1 Random Input

For Random Input, the bus was driven along a road at a constant speed (cruise control usually engaged) starting at 20 mph going in steps of 10 mph until 50 mph was reached and then from 50 mph back to 20 mph.

5.5.2 Step Input

The test bus was driven at constant speed over the Celanese Rail crossing at speeds of 20, 30, 40, and 50 mph while data from the transducers was recorded.

5.5.3 Well-Defined Input

With the bus drive and tag wheels on the four post shakers, various harmonic inputs were requested. (Communication was via remote headsets). Once the shaker posts began to operate, several seconds passed before recording started to allow transient vibrations to decay. The inputs were the following:

a) With all four posts in-phase and at an amplitude of ±1/8 inch, the following frequencies were used: (Hz) 0.5, 1.0, 1.5, 2.0, 2.5, 3.0, 4.0, 5.0, 6.0, 7.0, 7.5, 8.0, 8.5, 9.0, 9.1, 9.2, 9.3, 9.4, 9.5, 9.6, 9.7, 9.8, 9.9, 10.0, 11.0, 12.0, 13.0, 14.0 and 15.0.

b) With right and left shaker posts out-of-phase by 180° and at an amplitude of ±1/8 inch, the following frequencies were used: (Hz) 0.5, 0.8, 0.9, 1.0, 1.5, 2.0, 2.5, 3.0, 4.0, 5.0, 6.0, 7.0, 8.0, 9.0, 10.0, 11.0, 12.0, 13.0, 14.0 and 15.0.

Similar tests as above were run at different amplitudes (±0.15" and ±1/4") and with tag axles depressurized and with the torsion bar disconnected, more tests were run.

The harmonic tests coincided with Test Bus configurations #5, 6 and 7.

5.6 RESULTS

Typical results are plotted for body vertical acceleration at the c.g.; front axle vertical acceleration and drive axle vertical acceleration versus Time and Power Spectral Density plots of the same measured parameters versus frequency for random road ride quality input at 50 mph in bus configuration #3, #10 and #12. The graphs are presented in figures 5.5 - 5.16.

Time wave forms and PSD plots are shown in figures 5.17 - 5.22 for the same measured parameters as above with the bus in configuration #3 running over a step input (a railway crossing) at 50 mph.

Time wave forms are presented in figures 5.23 - 5.29 for vertical body accelerations within the bus (the c.g. location being the "middle sized" waveform) and at frequencies ranging from 1.4 Hz harmonic excitation to 2.0 Hz excitation. These graphs are given for bus configuration #5.

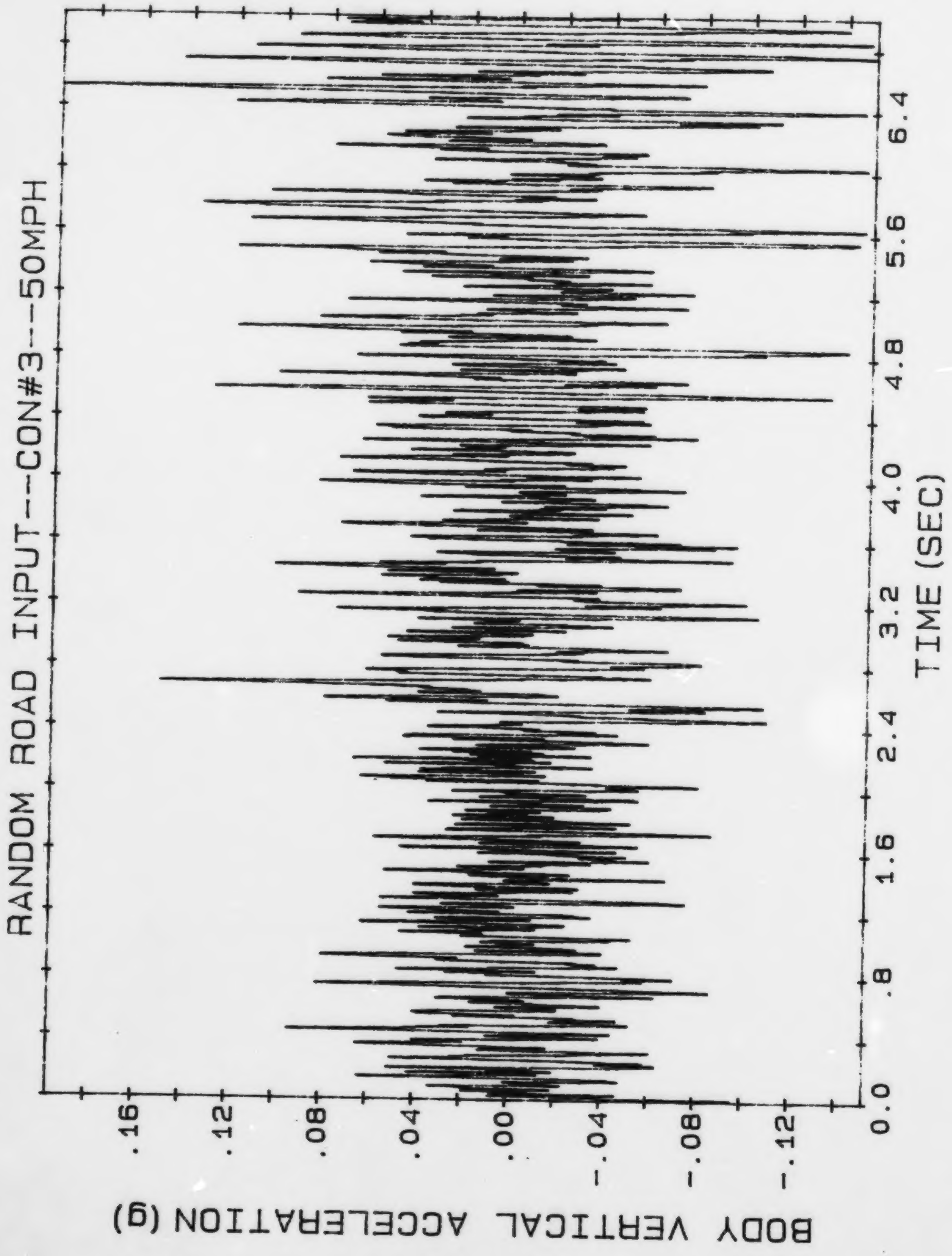

RANDOM ROAD INPUT---CON#3---50MPH
TIME (SEC)
0.0
.8
1.6
2.4
3.2
4.0
4.8
5.6
6.4
BODY VERTICAL ACCELERATION (g)
.16
.12
.08
.04
.00
-.04
-.08
-.12

FIGURE 5.5

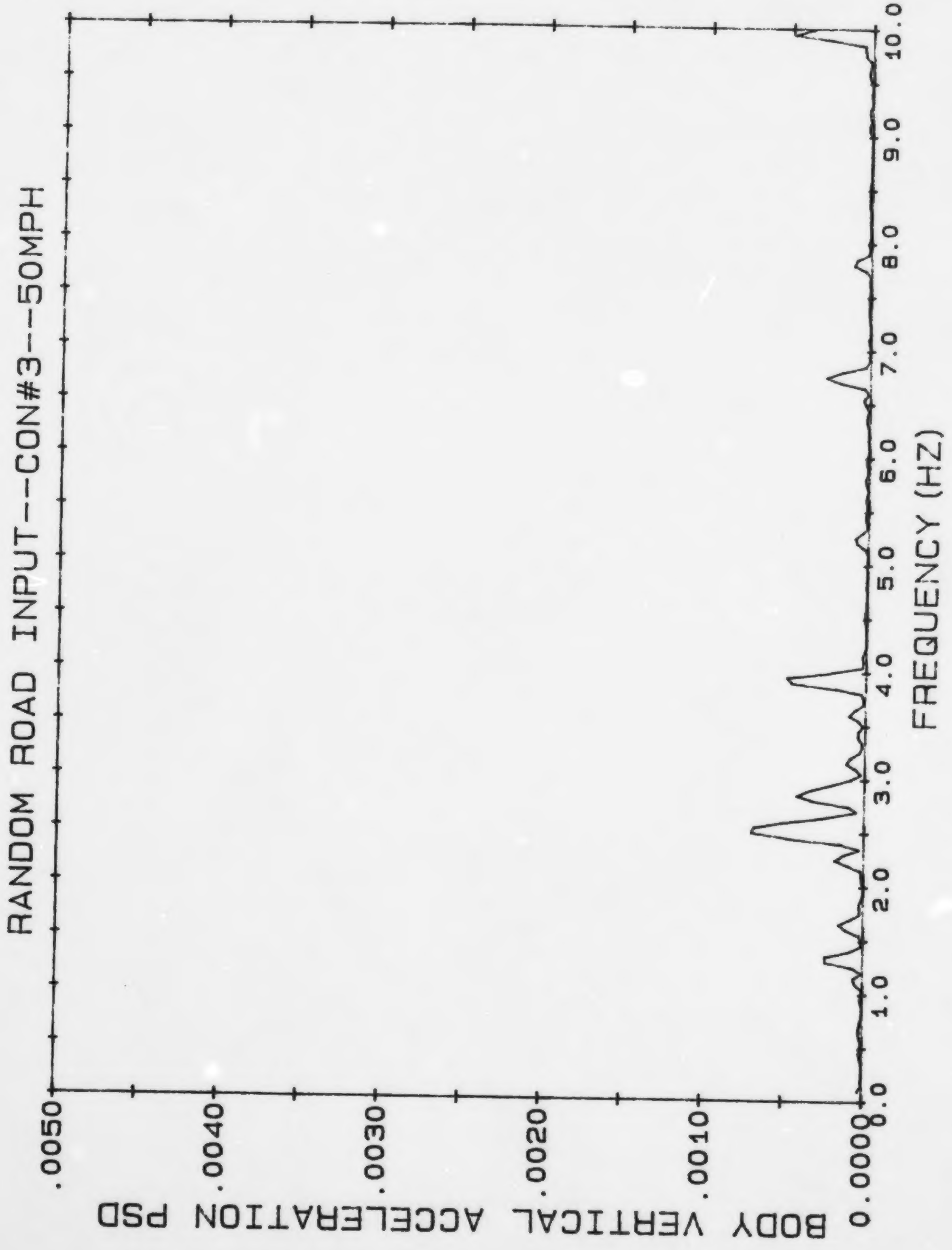

RANDOM ROAD INPUT--CON#3--50MPH
FREQUENCY (HZ)
0.0
1.0
2.0
3.0
4.0
5.0
6.0
7.0
8.0
9.0
10.0
BODY VERTICAL ACCELERATION PSD
0.0000
.0010
.0020
.0030
.0040
.0050

FIGURE 5.6

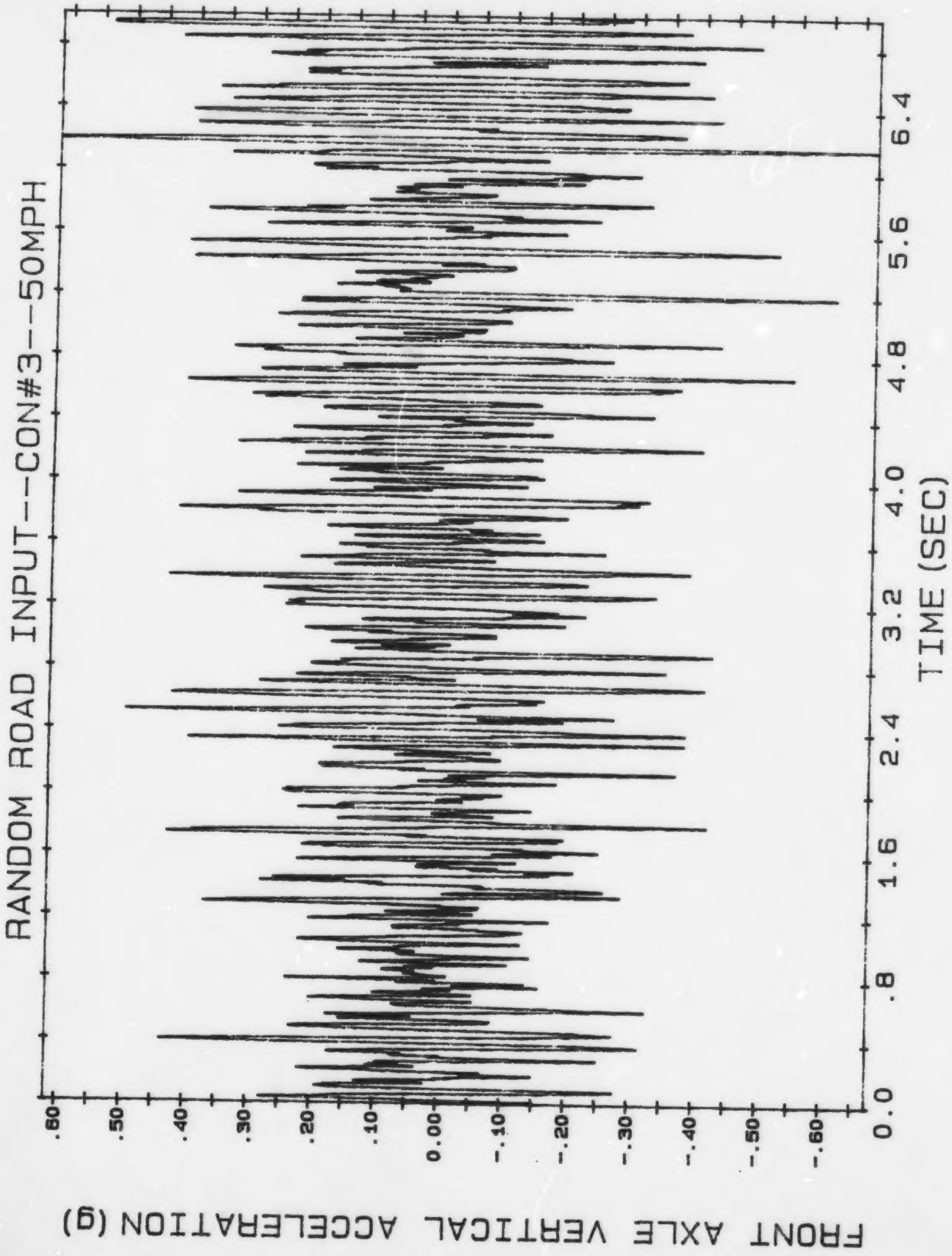

RANDOM ROAD INPUT--CON#3--50MPH
TIME (SEC)
0.0
.8
1.6
2.4
3.2
4.0
4.8
5.6
6.4
FRONT AXLE VERTICAL ACCELERATION (g)
.60
.50
.40
.30
.20
.10
0.00
-.10
-.20
-.30
-.40
-.50
-.60

FIGURE 5.7

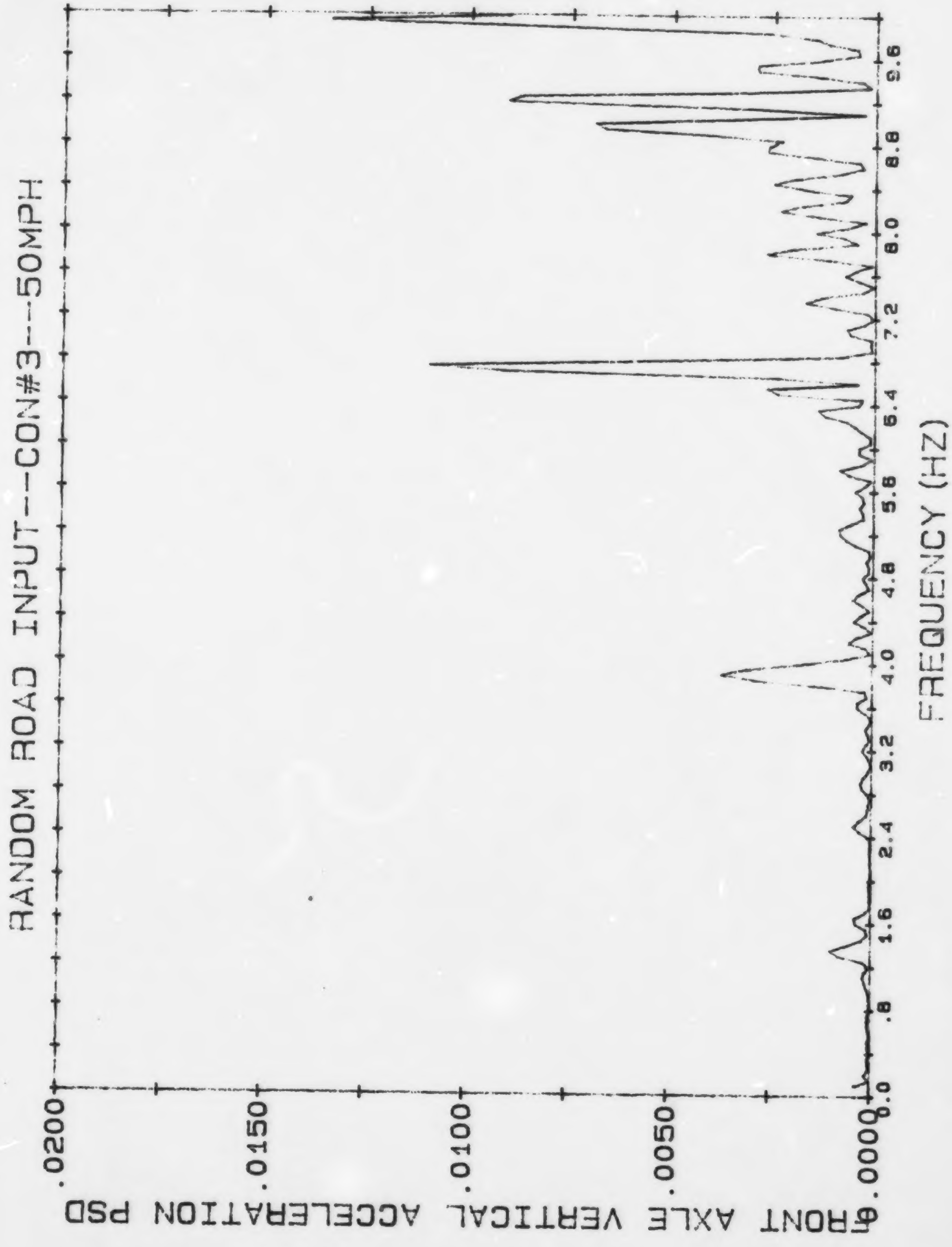

RANDOM ROAD INPUT---CON#3---50MPH
FREQUENCY (HZ)
0.0
.8
1.6
2.4
3.2
4.0
4.8
5.6
6.4
7.2
8.0
8.8
9.6
FRONT AXLE VERTICAL ACCELERATION PSD
.0000
.0050
.0100
.0150
.0200

FIGURE 5.8

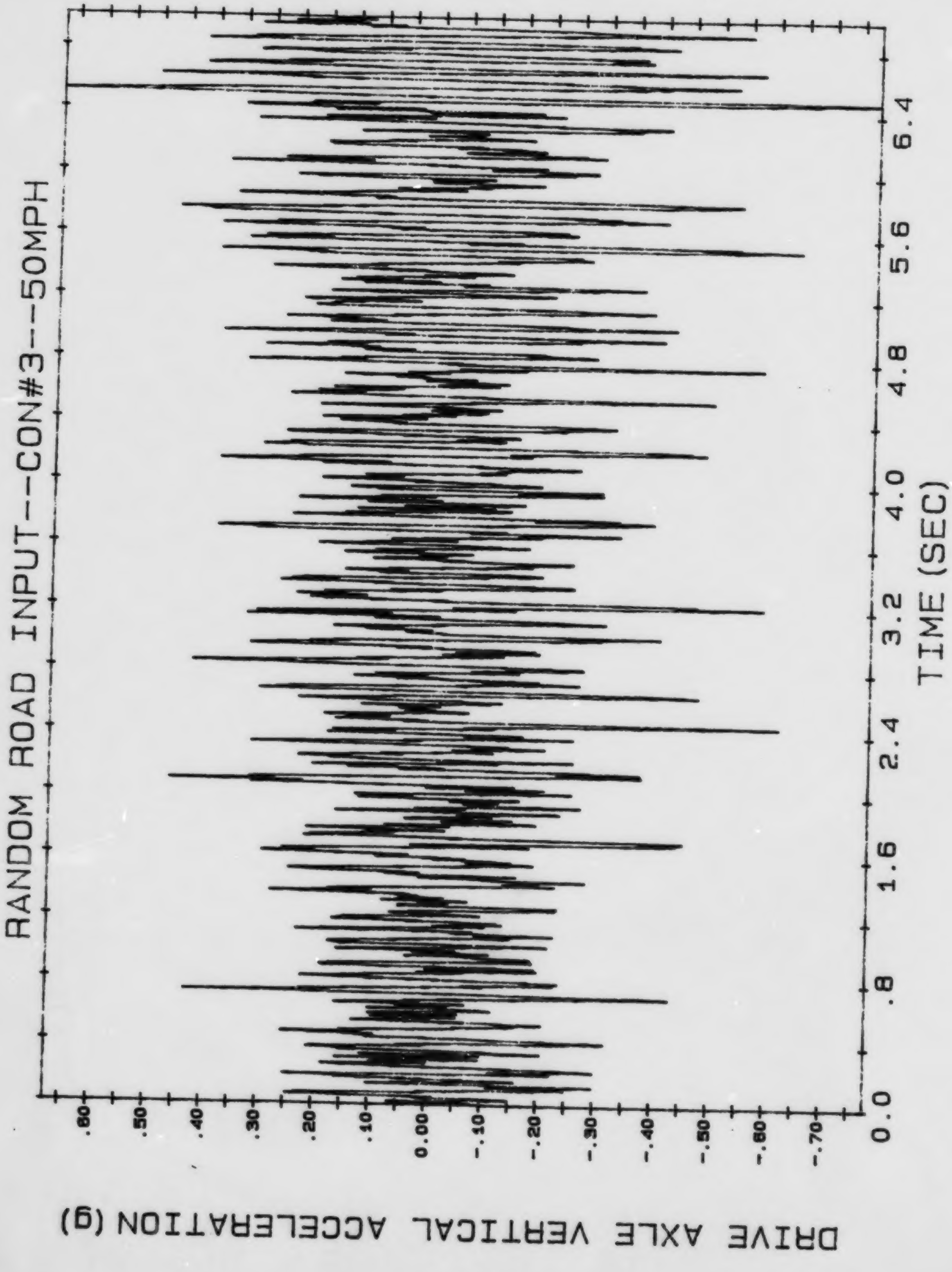

RANDOM ROAD INPUT--CON#3--50MPH
TIME (SEC)
0.0
.8
1.6
2.4
3.2
4.0
4.8
5.6
6.4
DRIVE AXLE VERTICAL ACCELERATION (g)
.60
.50
.40
.30
.20
.10
0.00
-.10
-.20
-.30
-.40
-.50
-.60
-.70

FIGURE 5.9

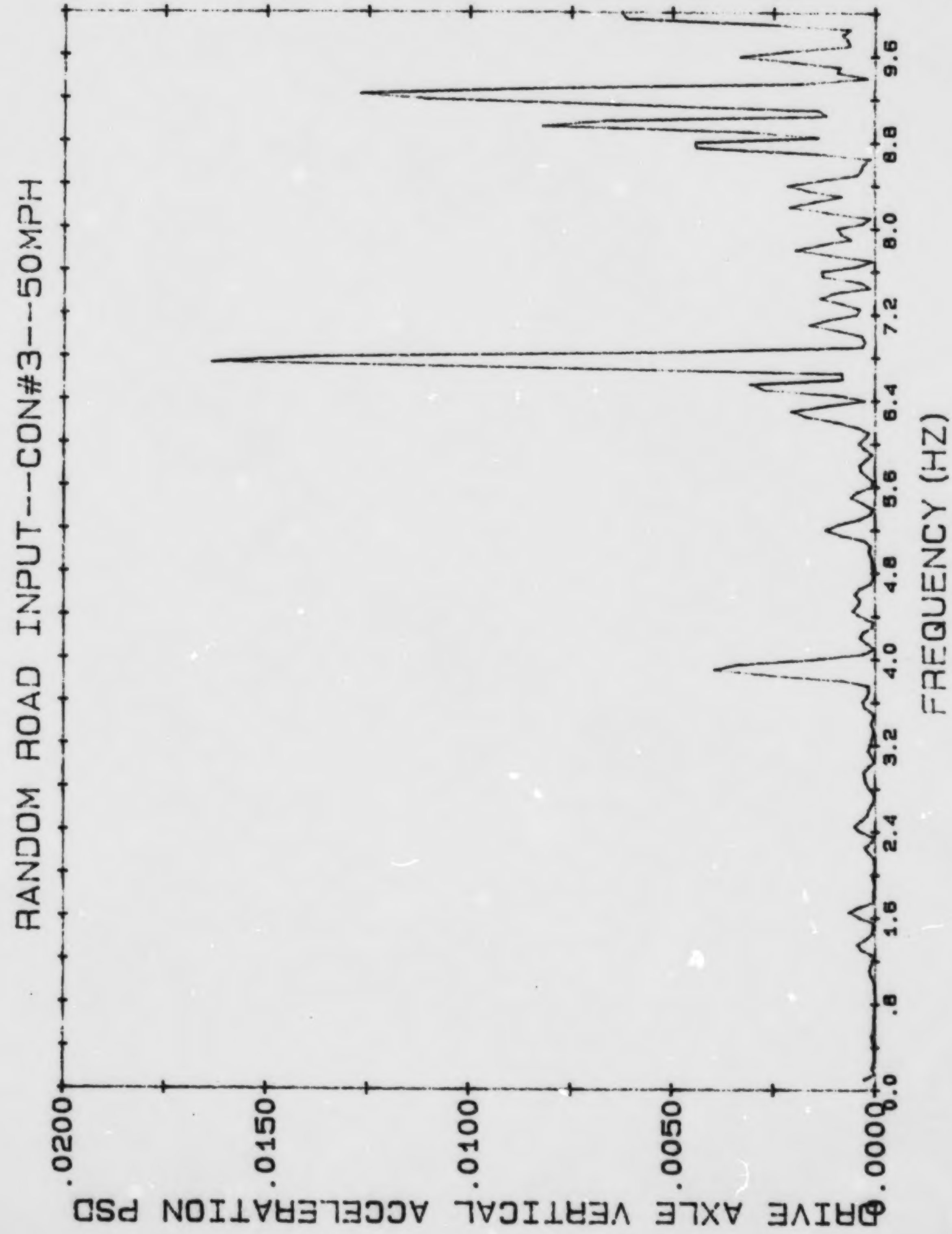

RANDOM ROAD INPUT--CON#3--50MPH
FREQUENCY (HZ)
0.0
.8
1.6
2.4
3.2
4.0
4.8
5.6
6.4
7.2
8.0
8.8
9.6
DRIVE AXLE VERTICAL ACCELERATION PSD
.0000
.0050
.0100
.0150
.0200

FIGURE 5.10

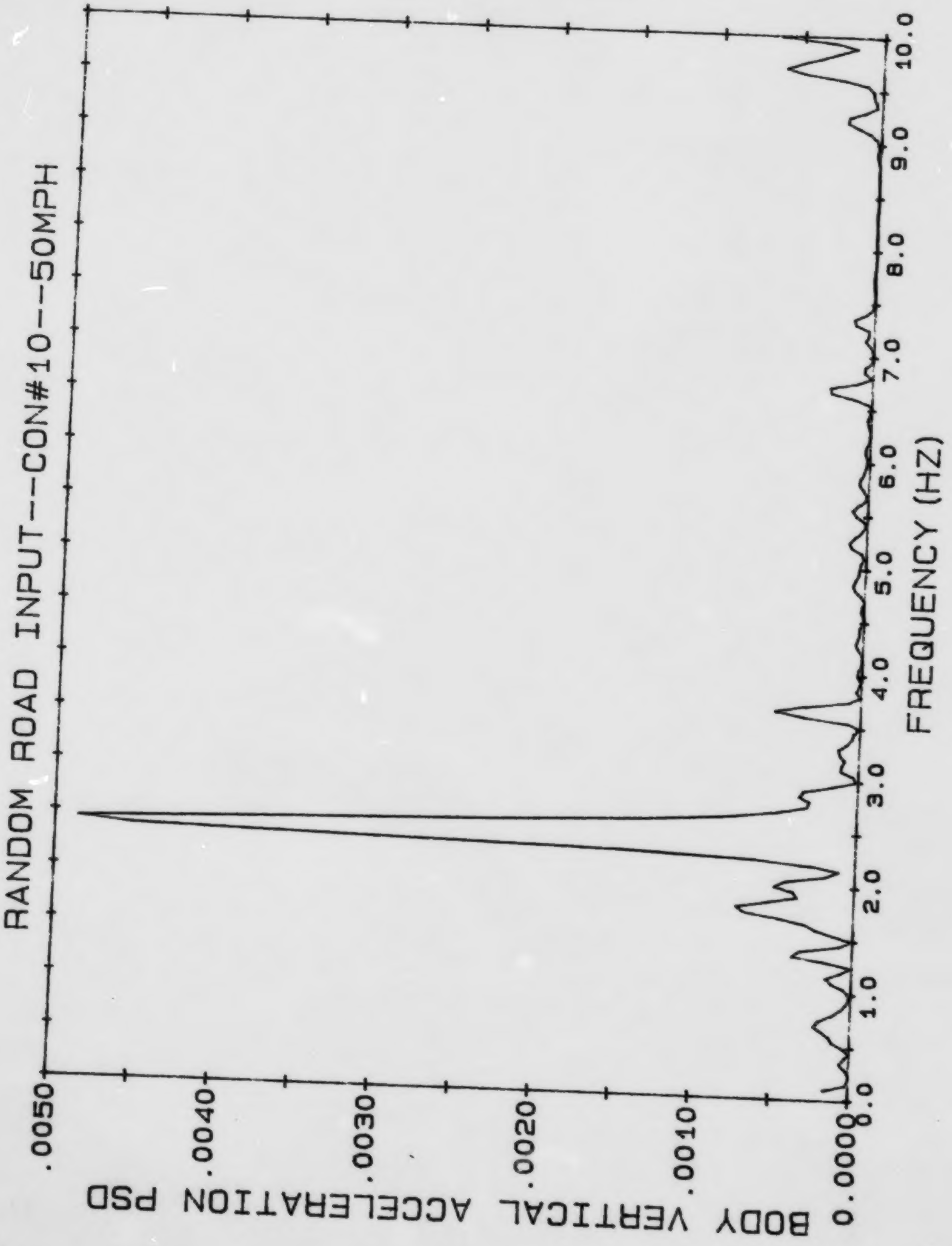

RANDOM ROAD INPUT--CON#10--50MPH
BODY VERTICAL ACCELERATION PSD
.0050
.0040
.0030
.0020
.0010
0.0000
0.0
1.0
2.0
3.0
4.0
5.0
6.0
7.0
8.0
9.0
10.0
FREQUENCY (HZ)

FIGURE 5.11

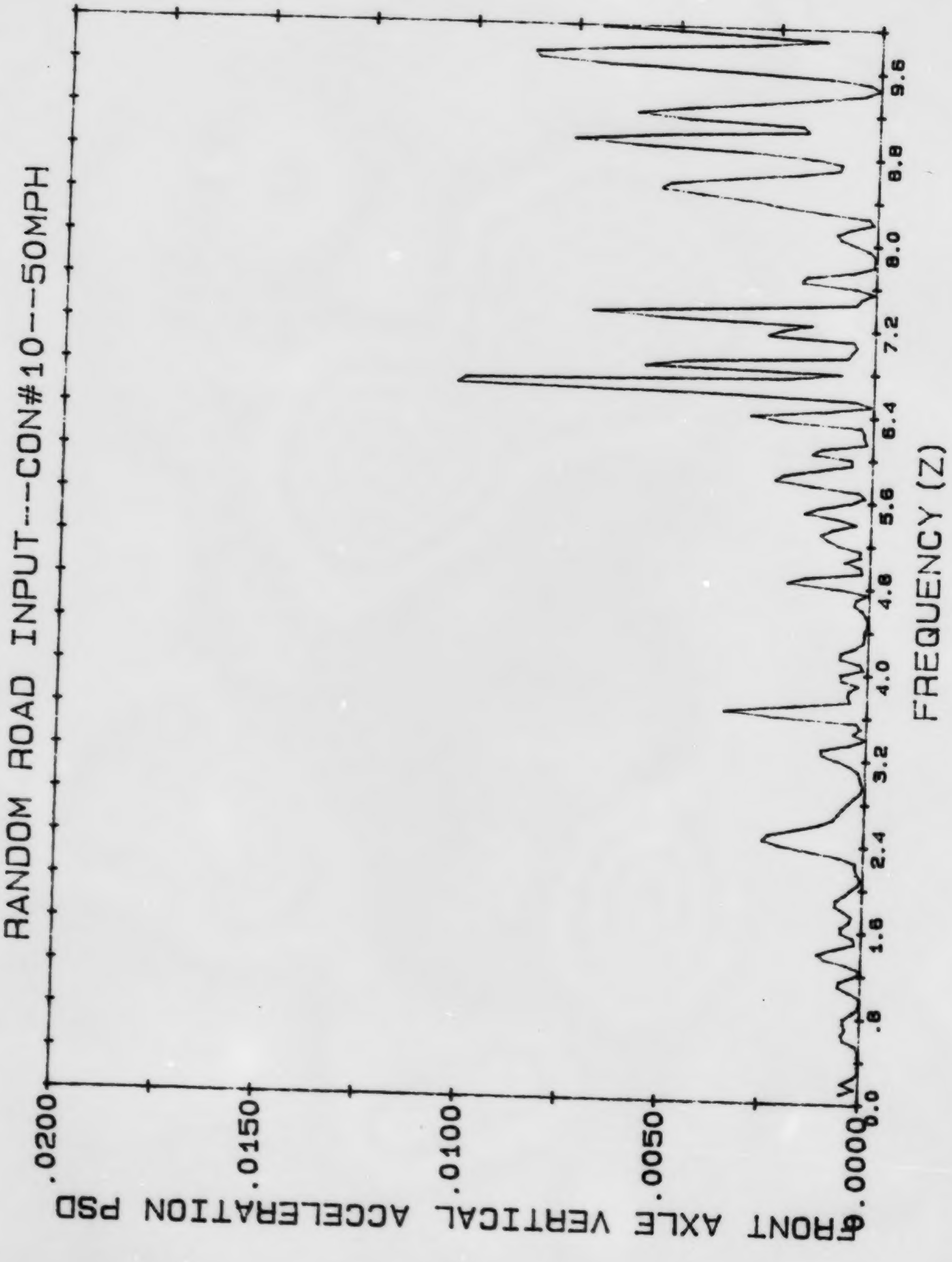

RANDOM ROAD INPUT---CON#10---50MPH
FREQUENCY (Z)
0.0
.8
1.6
2.4
3.2
4.0
4.8
5.6
6.4
7.2
8.0
8.8
9.6
FRONT AXLE VERTICAL ACCELERATION PSD
.0000
.0050
.0100
.0150
.0200

FIGURE 5.12

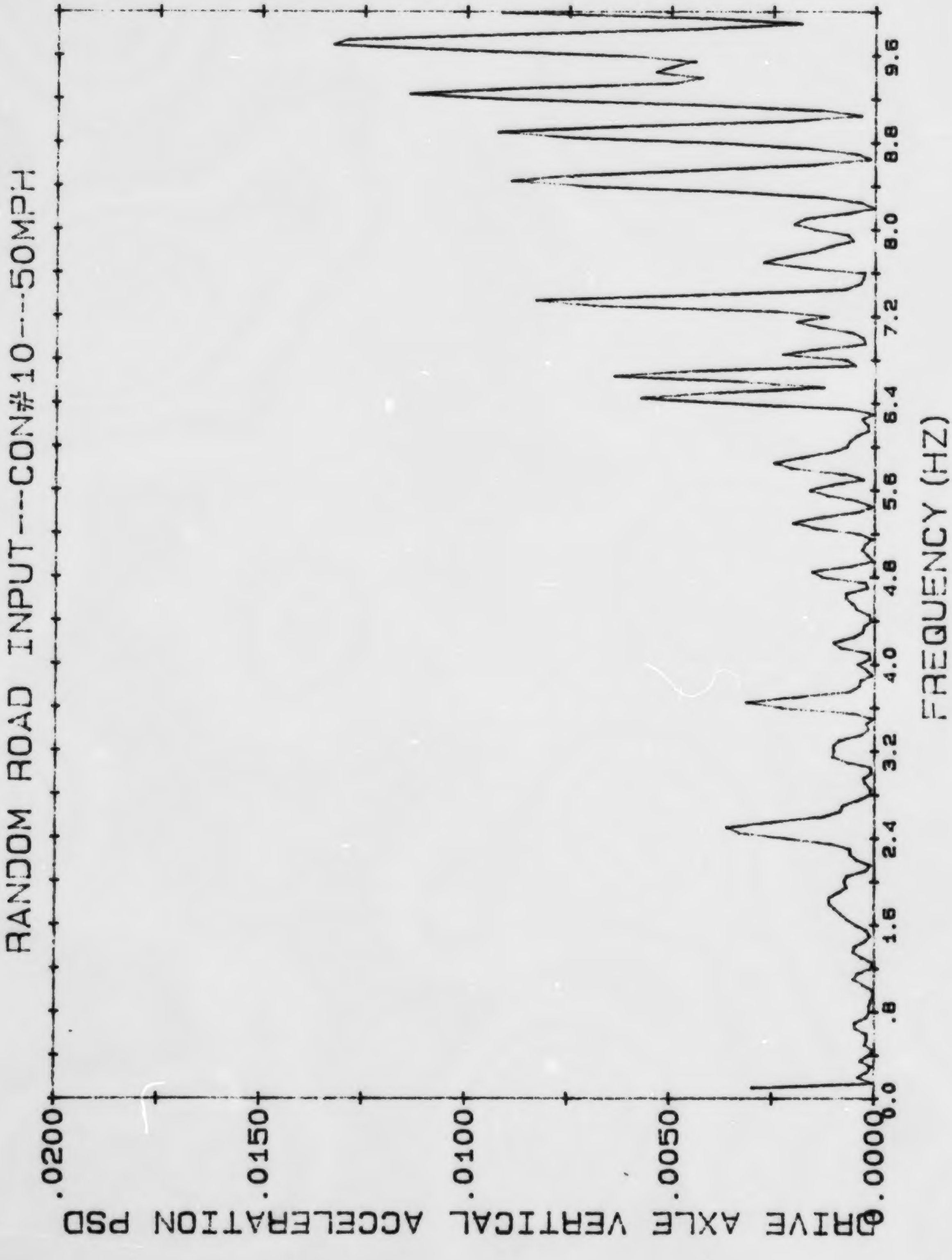

RANDOM ROAD INPUT---CON#10---50MPH
DRIVE AXLE VERTICAL ACCELERATION PSD
.0200
.0150
.0100
.0050
.0000
FREQUENCY (HZ)
0.0
.8
1.6
2.4
3.2
4.0
4.8
5.6
6.4
7.2
8.0
8.8
9.6

FIGURE 5.13

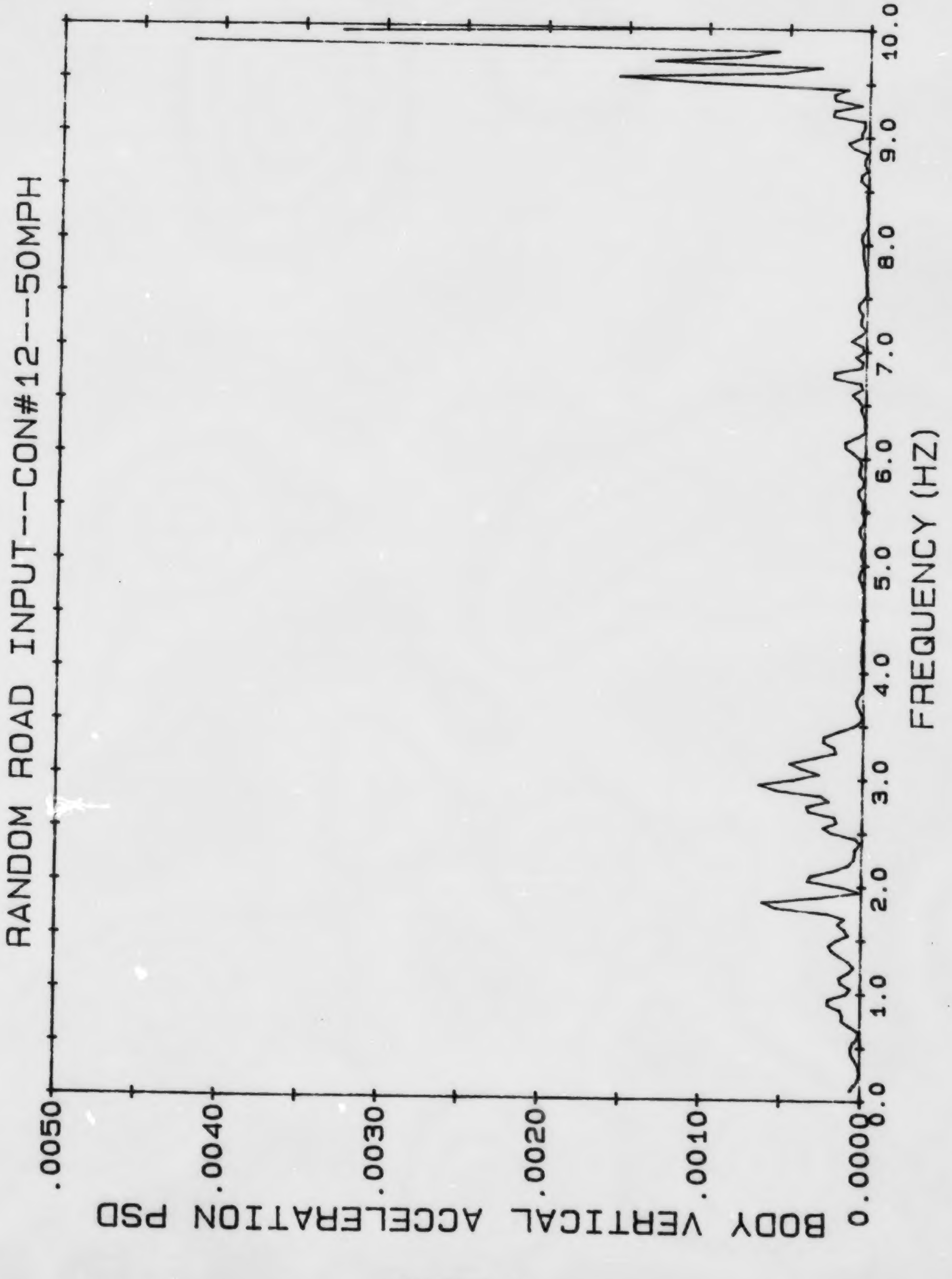

RANDOM ROAD INPUT--CON#12--50MPH
BODY VERTICAL ACCELERATION PSD
.0050
.0040
.0030
.0020
.0010
0.0000
FREQUENCY (HZ)
0.0
1.0
2.0
3.0
4.0
5.0
6.0
7.0
8.0
9.0
10.0

FIGURE 5.14

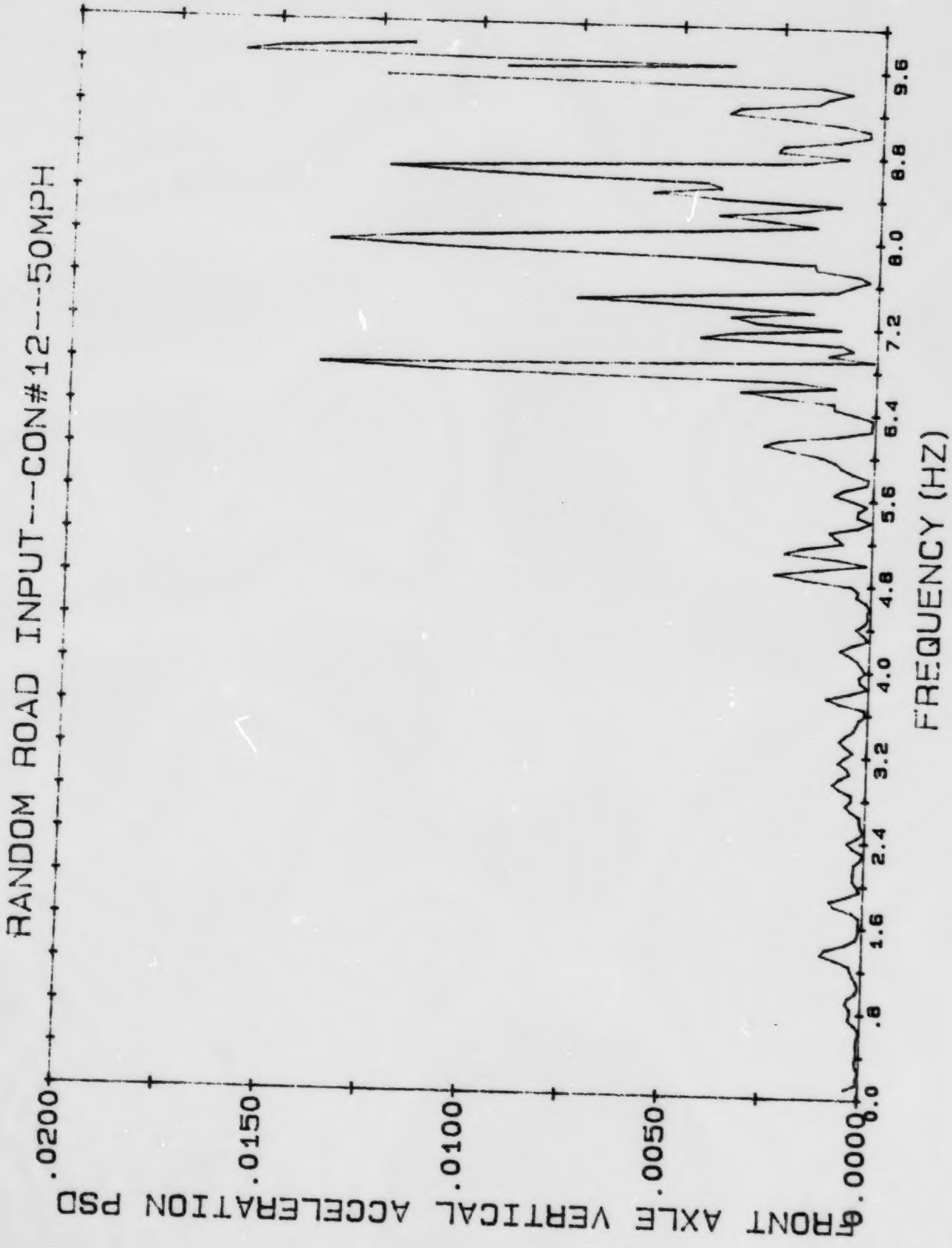

RANDOM ROAD INPUT---CON#12---50MPH
FRONT AXLE VERTICAL ACCELERATION PSD
FREQUENCY (HZ)
.0000
.0050
.0100
.0150
.0200
0.0
.8
1.6
2.4
3.2
4.0
4.8
5.6
6.4
7.2
8.0
8.8
9.6

FIGURE 5.15

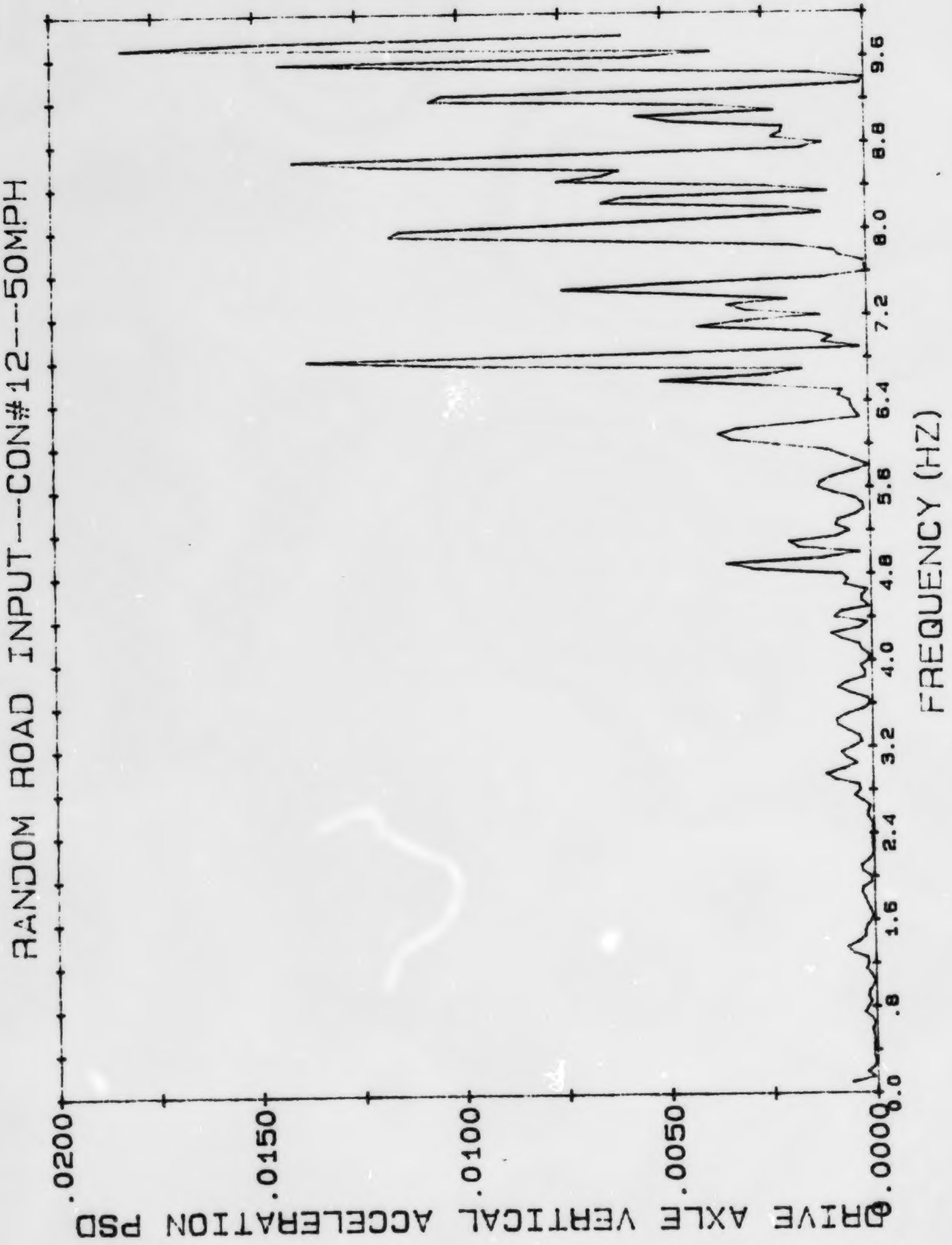

RANDOM ROAD INPUT---CON#12---50MPH
DRIVE AXLE VERTICAL ACCELERATION PSD
.0200
.0150
.0100
.0050
0.0000
FREQUENCY (HZ)
0.0
.8
1.6
2.4
3.2
4.0
4.8
5.6
6.4
7.2
8.0
8.8
9.6

FIGURE 5.16

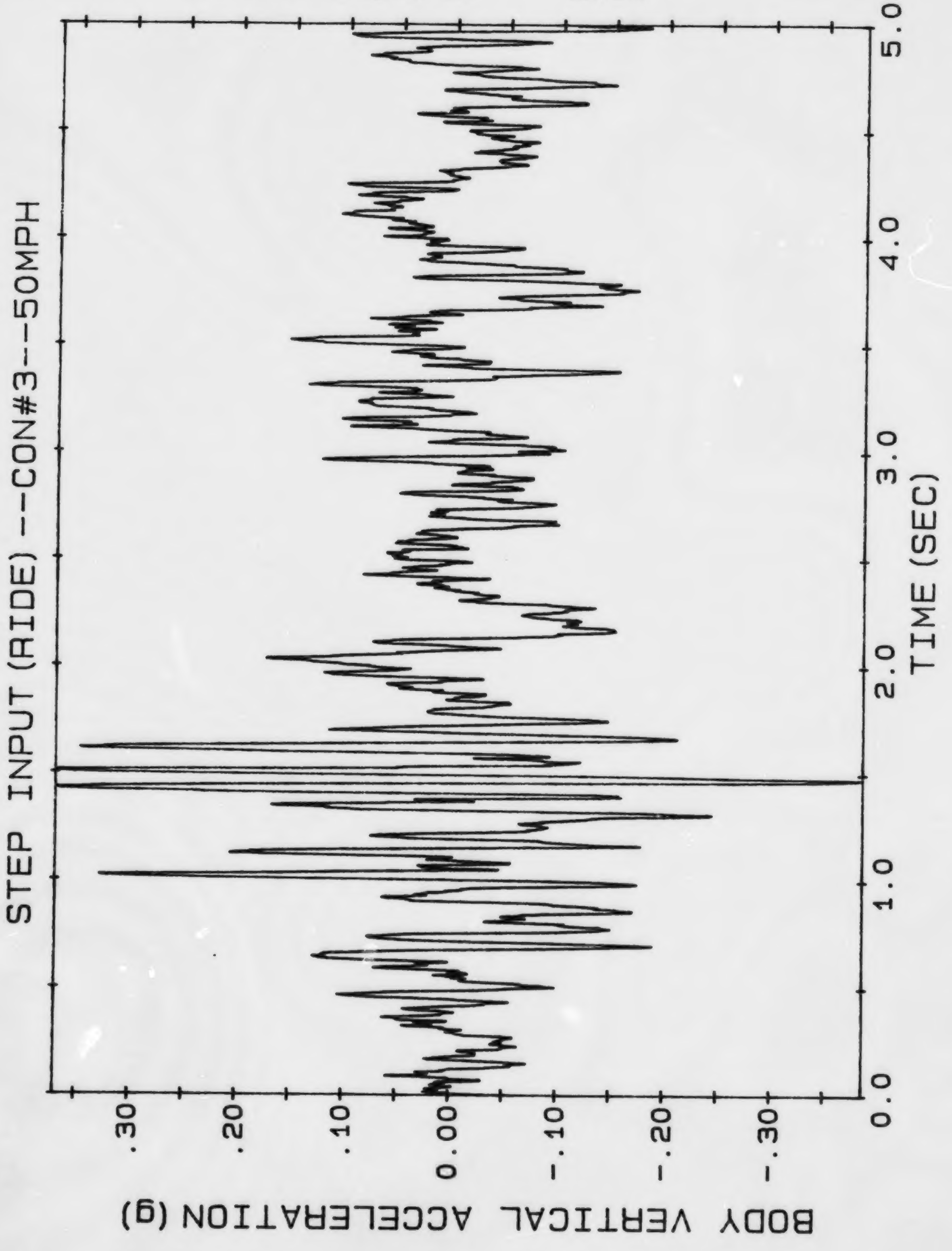

STEP INPUT (RIDE) --CON#3--50MPH
BODY VERTICAL ACCELERATION (g)
.30
.20
.10
0.00
-.10
-.20
-.30
TIME (SEC)
0.0
1.0
2.0
3.0
4.0
5.0

FIGURE 5.17

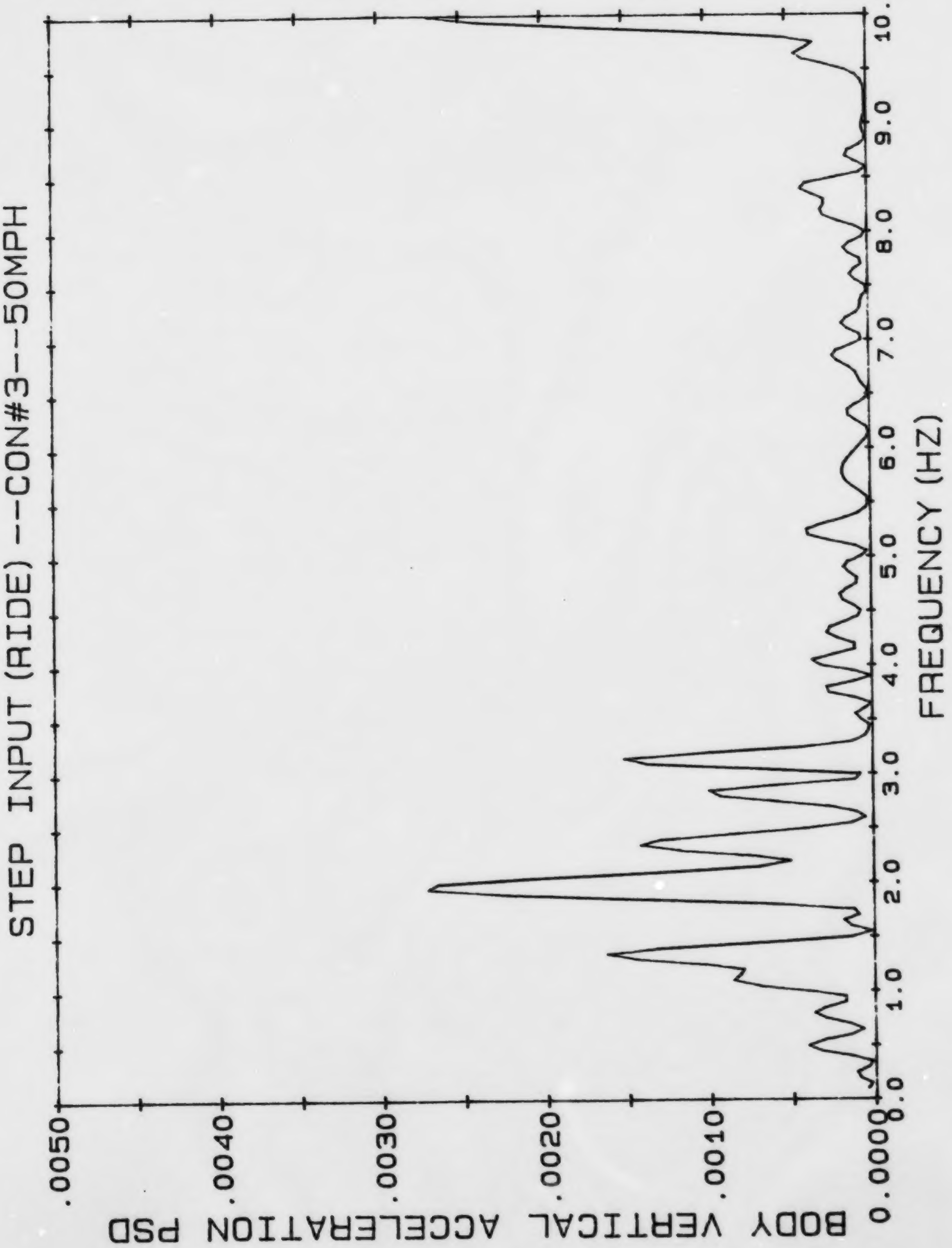

STEP INPUT (RIDE) --CON#3--50MPH
FREQUENCY (HZ)
BODY VERTICAL ACCELERATION PSD
0.0
1.0
2.0
3.0
4.0
5.0
6.0
7.0
8.0
9.0
10.0
.0050
.0040
.0030
.0020
.0010
0.0000

FIGURE 5.18

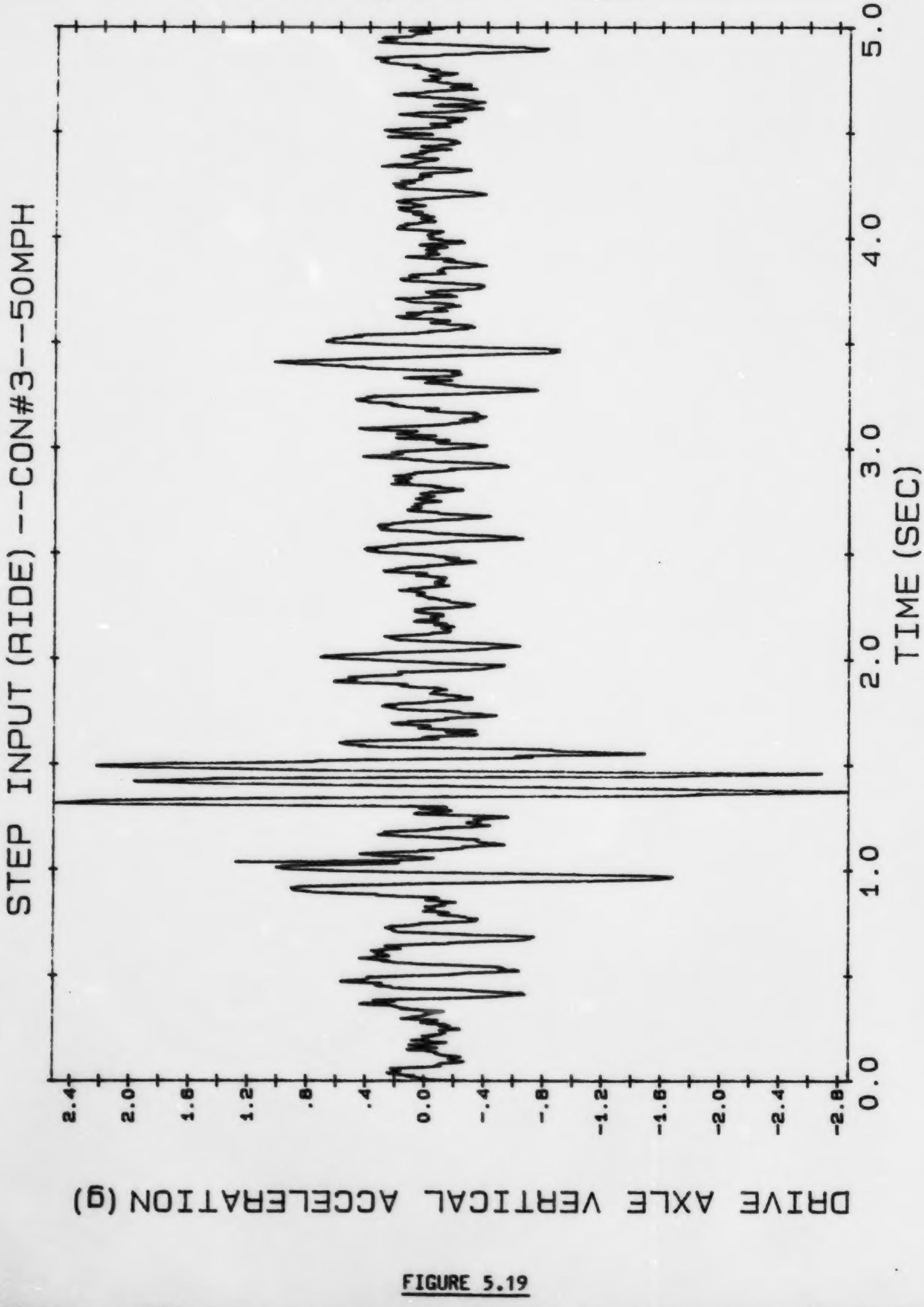

STEP INPUT (RIDE) --CON#3--50MPH
TIME (SEC)
0.0
1.0
2.0
3.0
4.0
5.0
DRIVE AXLE VERTICAL ACCELERATION (g)
2.4
2.0
1.6
1.2
.8
.4
0.0
-.4
-.8
-1.2
-1.6
-2.0
-2.4
-2.8

FIGURE 5.19

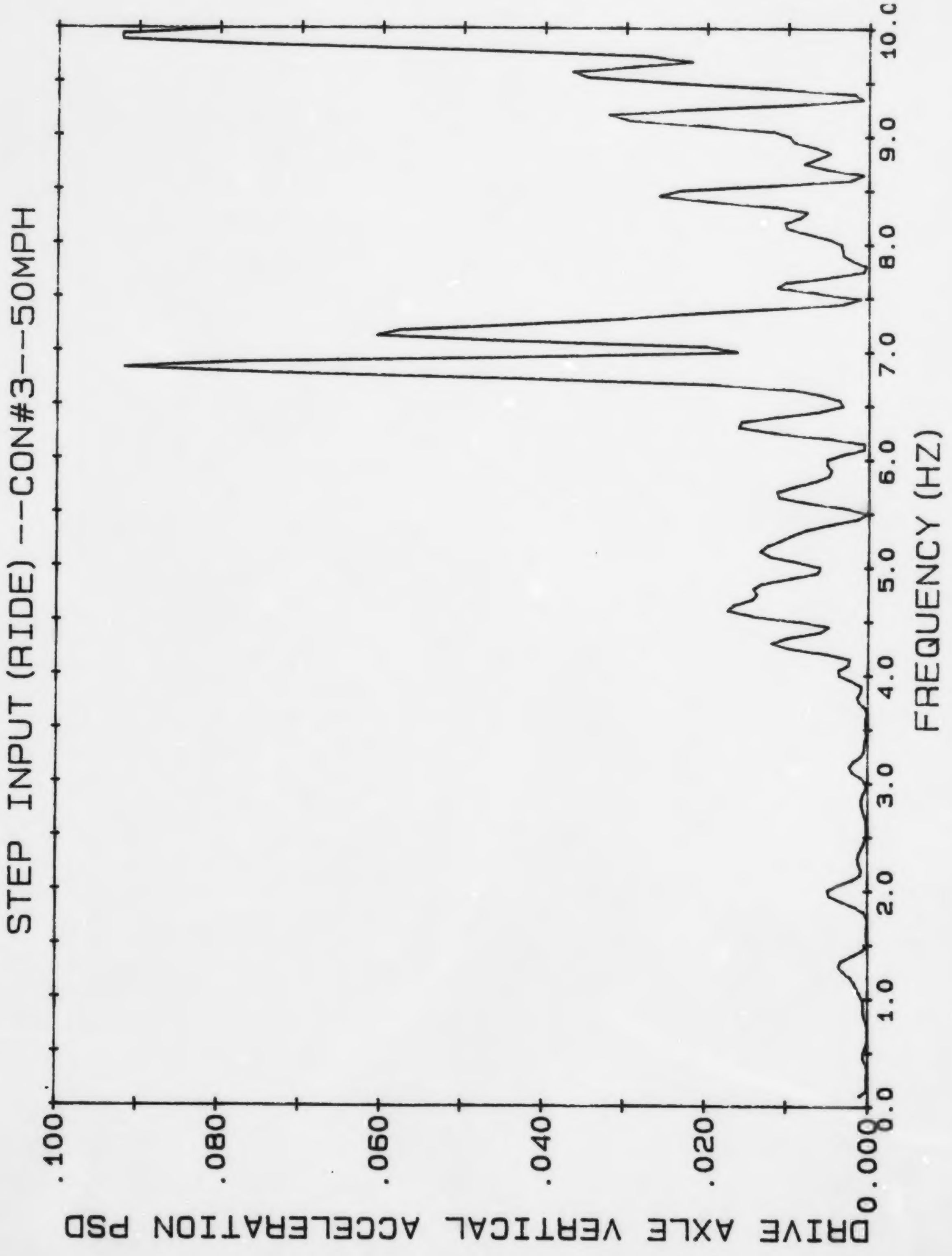

STEP INPUT (RIDE) --CON#3--50MPH
FREQUENCY (HZ)
0.0
1.0
2.0
3.0
4.0
5.0
6.0
7.0
8.0
9.0
10.0
DRIVE AXLE VERTICAL ACCELERATION PSD
.100
.080
.060
.040
.020
0.000

FIGURE 5.20

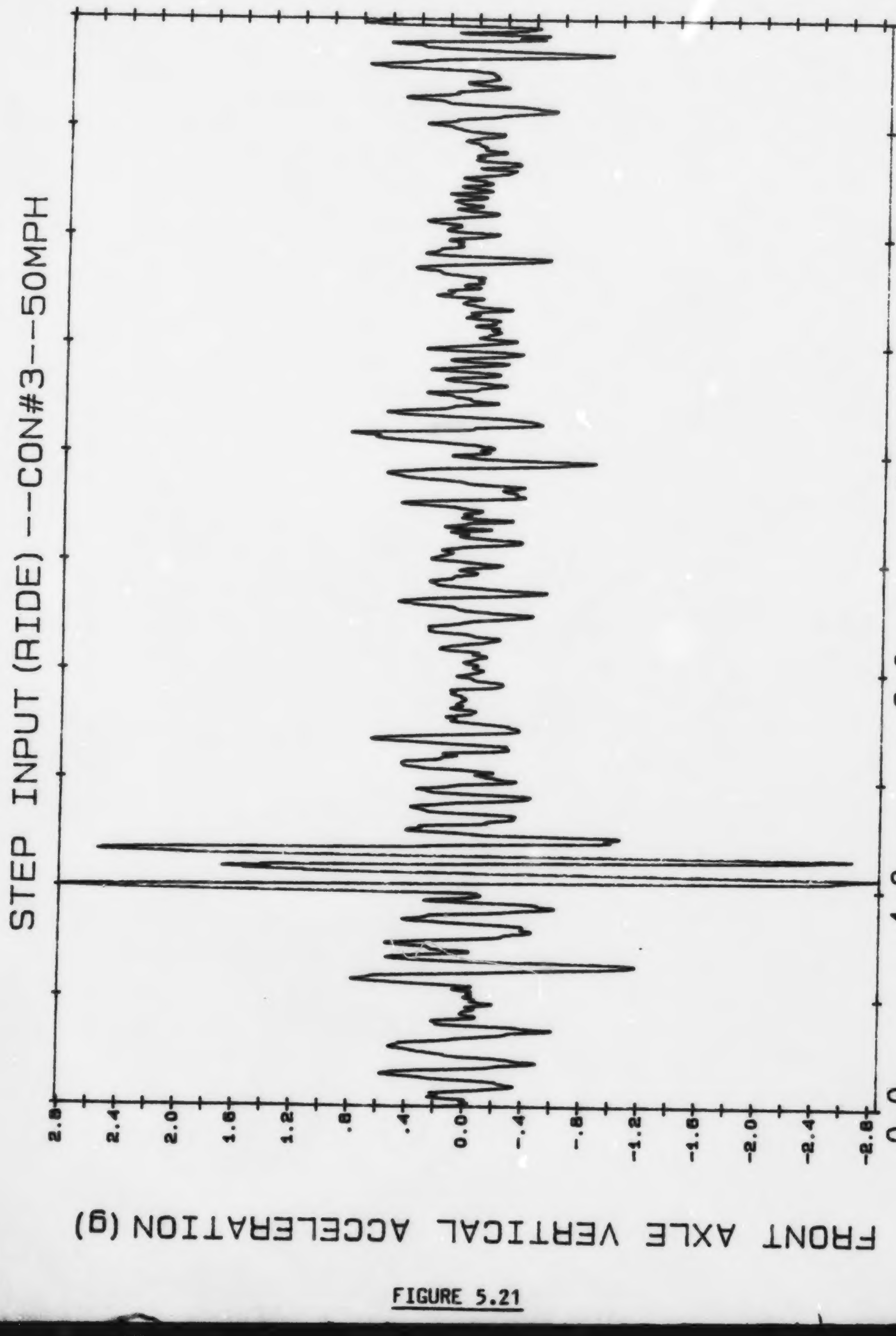

STEP INPUT (RIDE) --CON#3--50MPH
TIME (SEC)
0.0
1.0
2.0
3.0
4.0
5.0
FRONT AXLE VERTICAL ACCELERATION (g)
2.8
2.4
2.0
1.6
1.2
.8
.4
0.0
-.4
-.8
-1.2
-1.6
-2.0
-2.4
-2.8

FIGURE 5.21

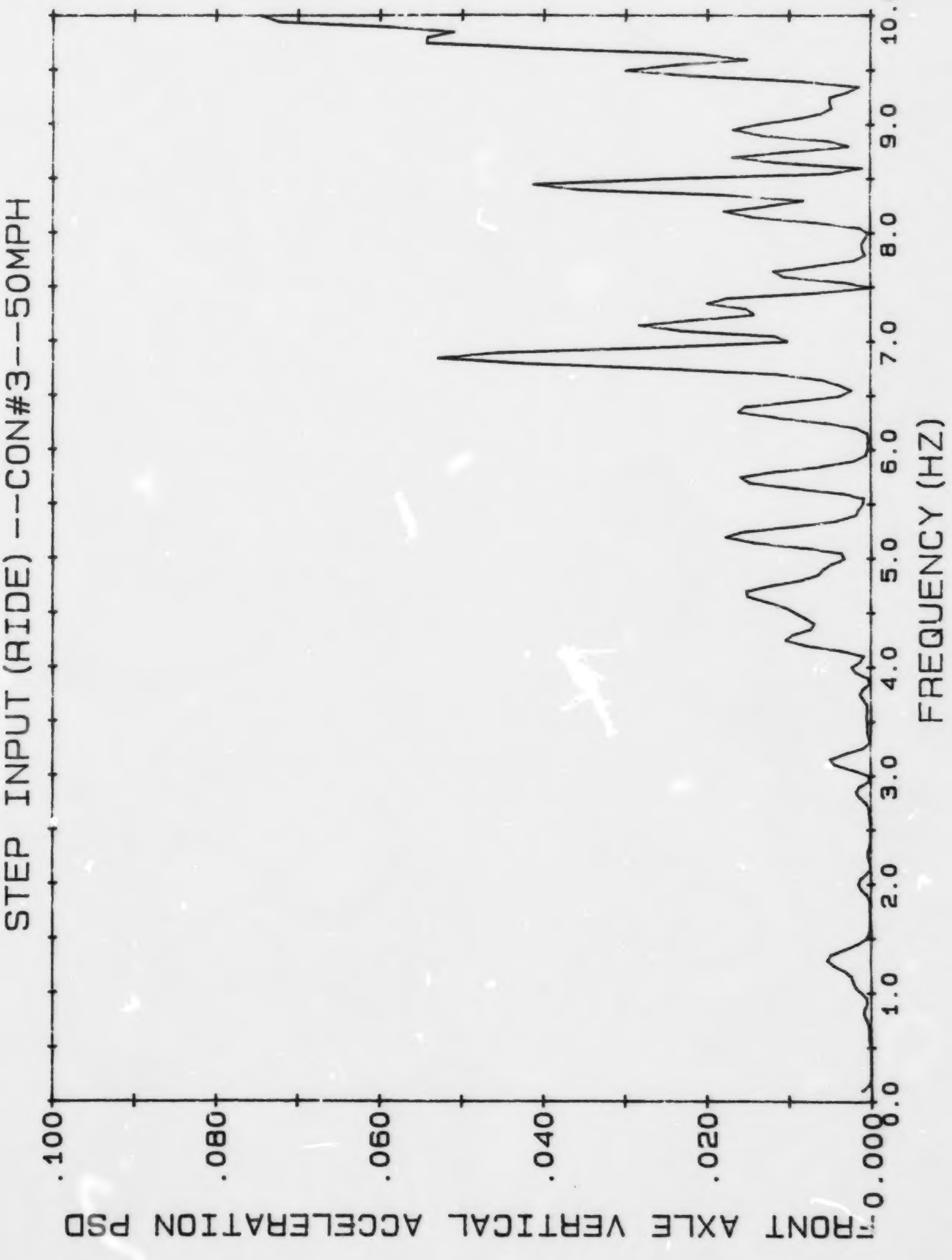

STEP INPUT (RIDE) --CON#3--50MPH
FRONT AXLE VERTICAL ACCELERATION PSD
FREQUENCY (HZ)
0.0
1.0
2.0
3.0
4.0
5.0
6.0
7.0
8.0
9.0
10.0
.100
.080
.060
.040
.020
0.000

FIGURE 5.22

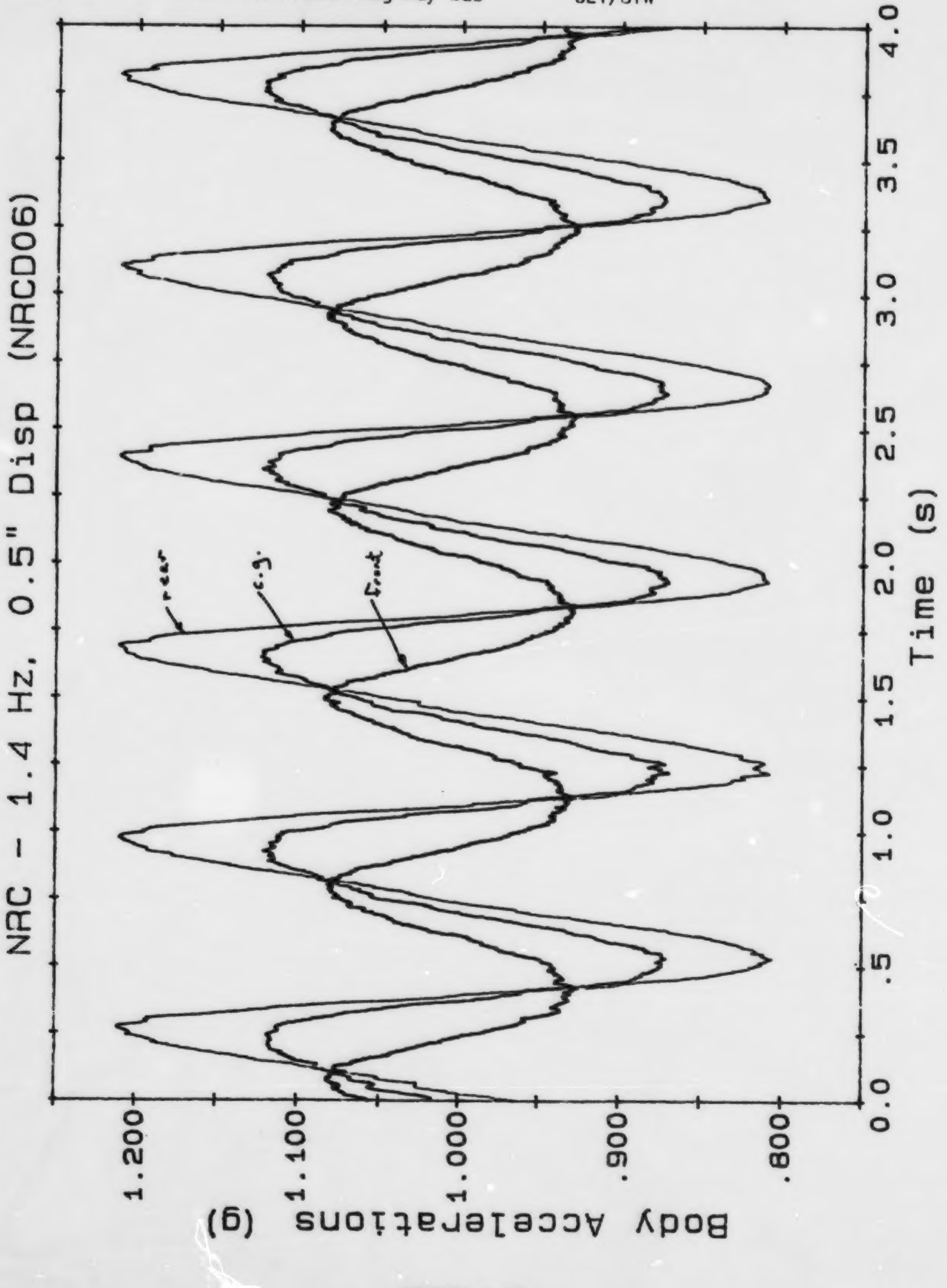

NRC - 1.4 Hz. 0.5" Disp (NRCD06)
Body Accelerations (g)
1.200
1.100
1.000
.900
.800
Time (s)
0.0
.5
1.0
1.5
2.0
2.5
3.0
3.5
4.0
rear
c.g.
front

FIGURE 5.23

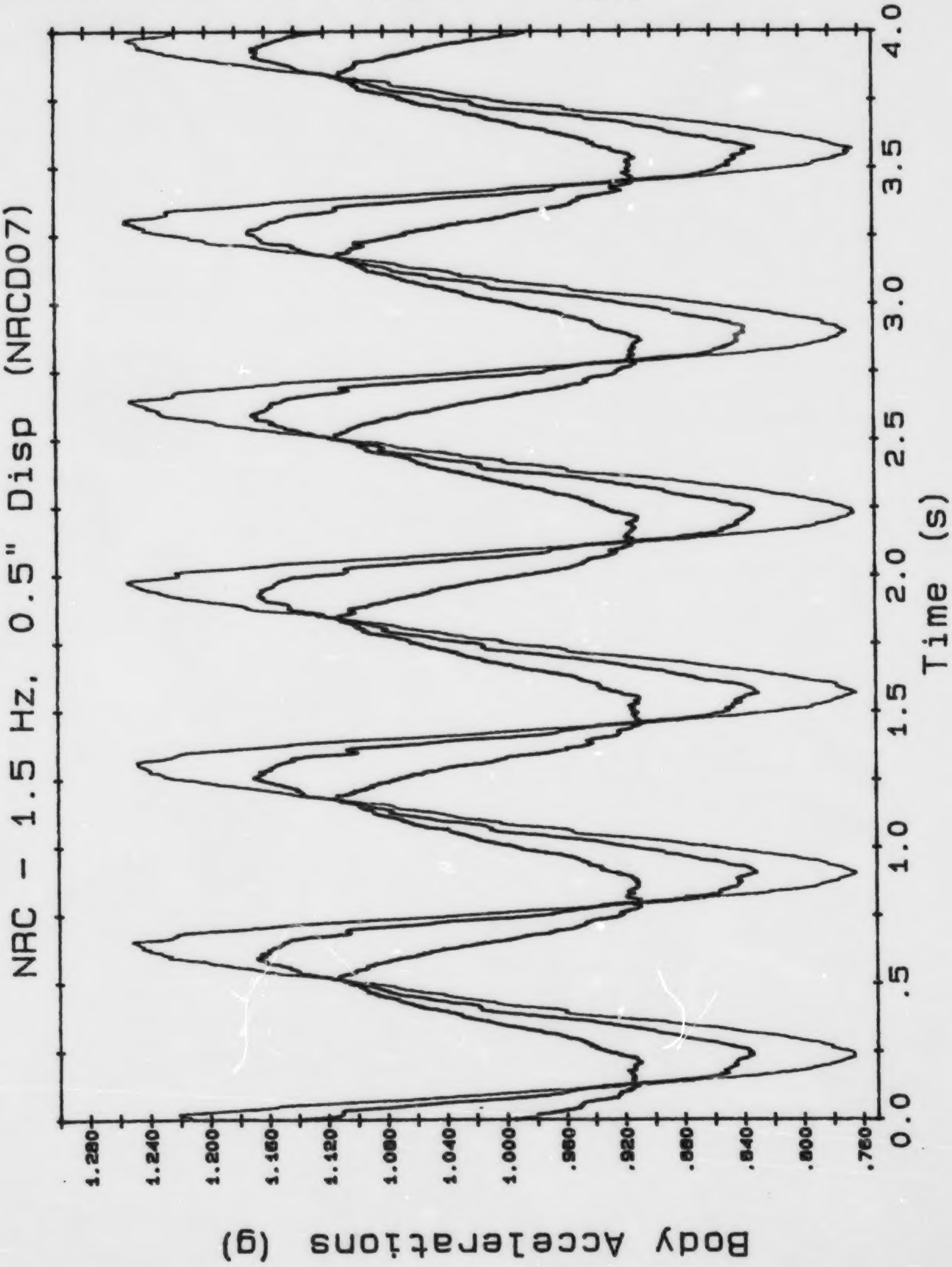

NRC - 1.5 Hz, 0.5" Disp (NRCD07)
Body Accelerations (g)
Time (s)
1.280
1.240
1.200
1.160
1.120
1.080
1.040
1.000
.960
.920
.880
.840
.800
.760
0.0
.5
1.0
1.5
2.0
2.5
3.0
3.5
4.0

FIGURE 5.24

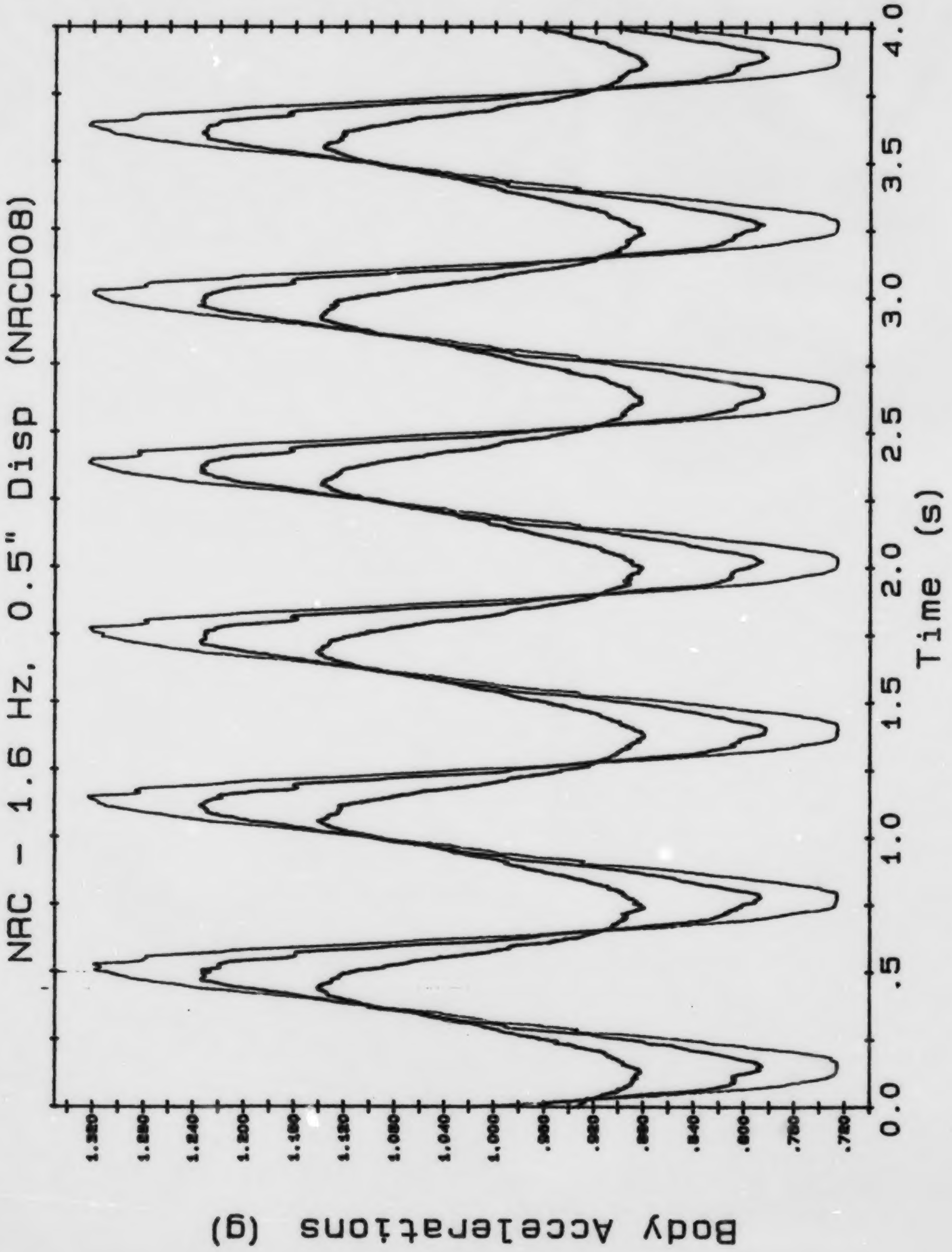

NRC - 1.6 Hz. 0.5" Disp (NRCD08)
Body Accelerations (g)
1.320
1.280
1.240
1.200
1.160
1.120
1.080
1.040
1.000
.960
.920
.880
.840
.800
.760
.720
Time (s)
0.0
.5
1.0
1.5
2.0
2.5
3.0
3.5
4.0

FIGURE 5.25

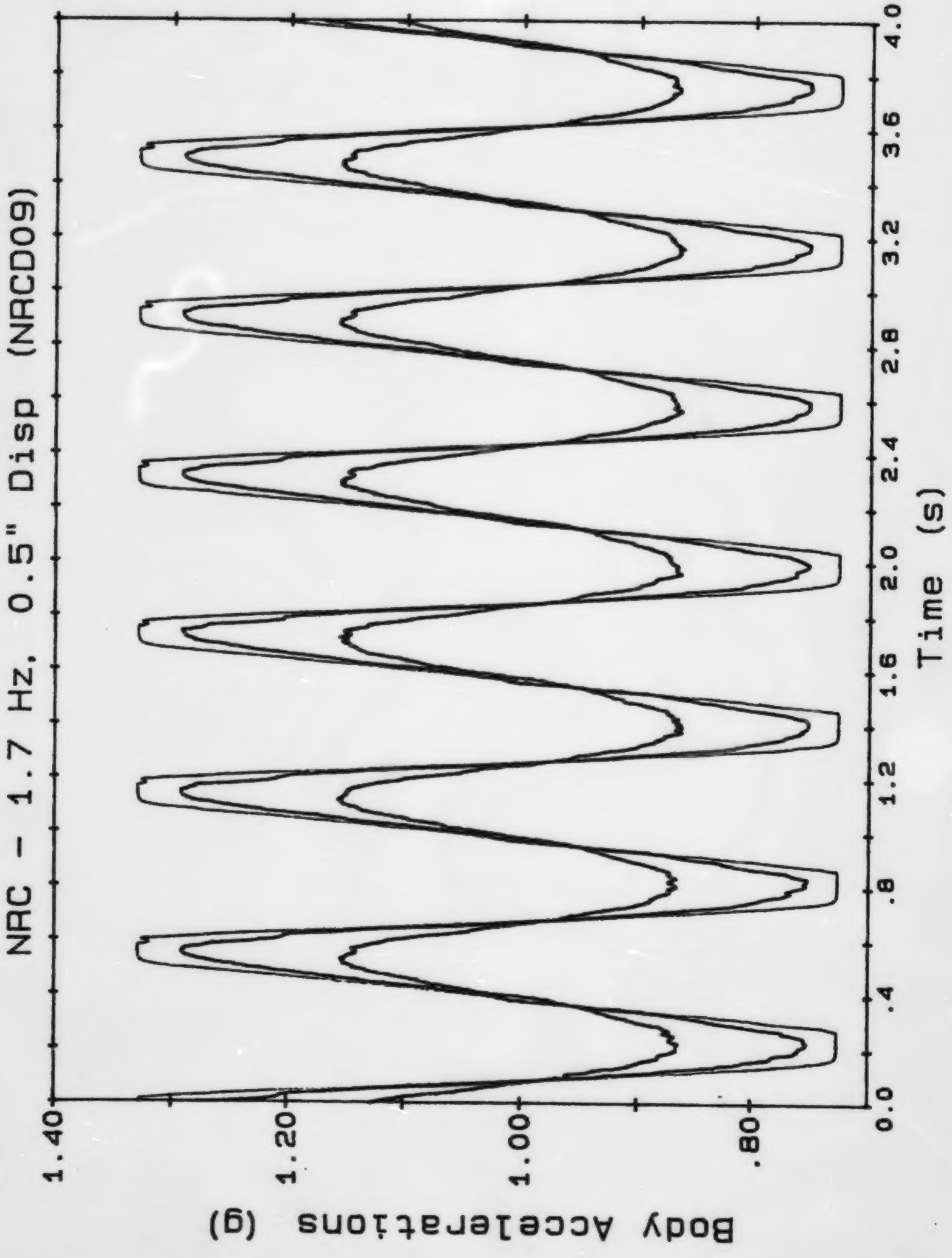

NRC - 1.7 Hz. 0.5" Disp (NRCD09)
Body Accelerations (g)
1.40
1.20
1.00
.80
Time (s)
0.0
.4
.8
1.2
1.6
2.0
2.4
2.8
3.2
3.6
4.0

FIGURE 5.26

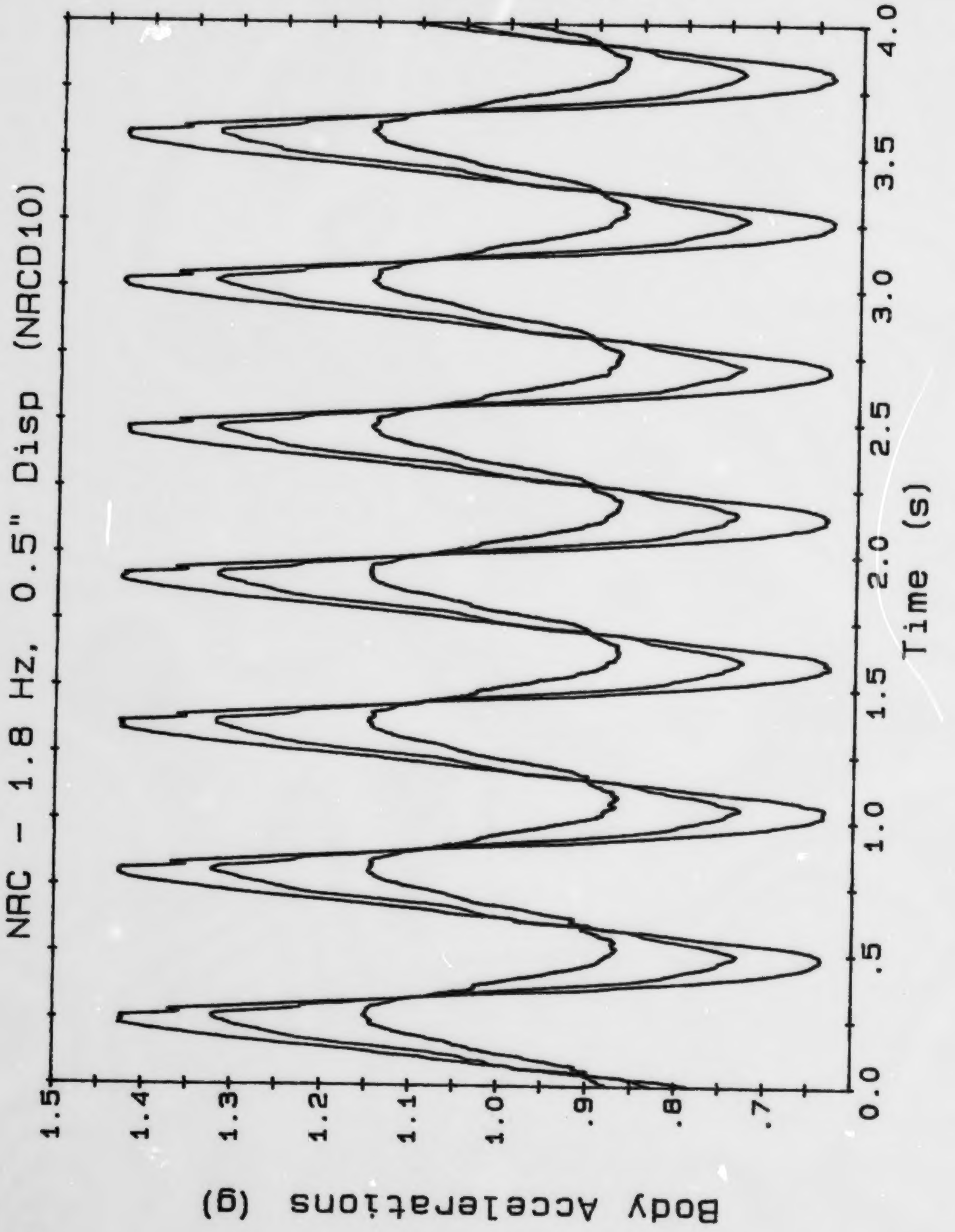

NRC - 1.8 Hz. 0.5" Disp (NRCD10)
Body Accelerations (g)
Time (s)
1.5
1.4
1.3
1.2
1.1
1.0
.9
.8
.7
0.0
.5
1.0
1.5
2.0
2.5
3.0
3.5
4.0

FIGURE 5.27

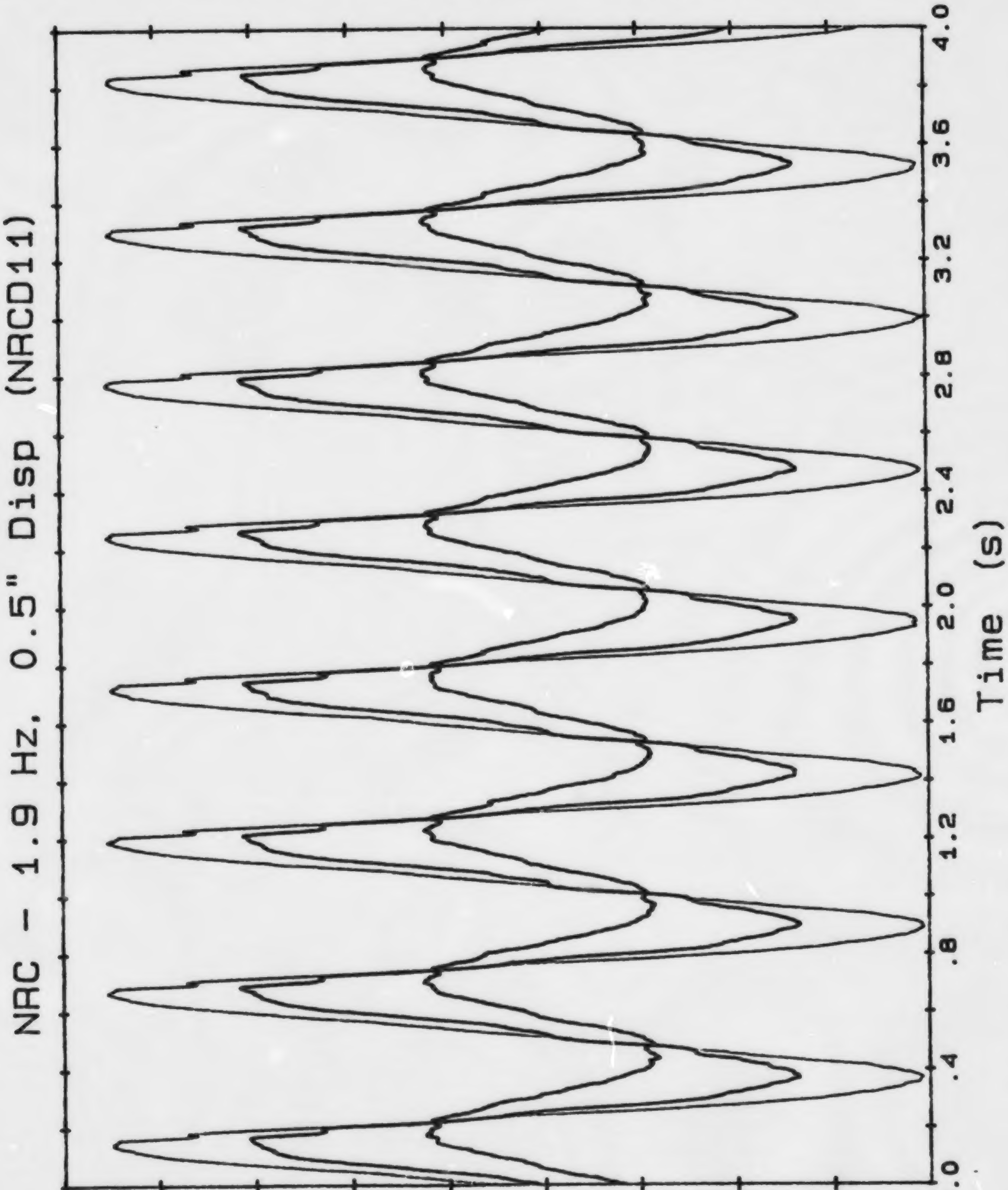

NRC - 1.9 Hz. 0.5" Disp (NRCD11)
Time (s)
0.0
.4
.8
1.2
1.6
2.0
2.4
2.8
3.2
3.6
4.0
Body Acceleration (g)
1.40
1.20
1.00
.80

FIGURE 5.28

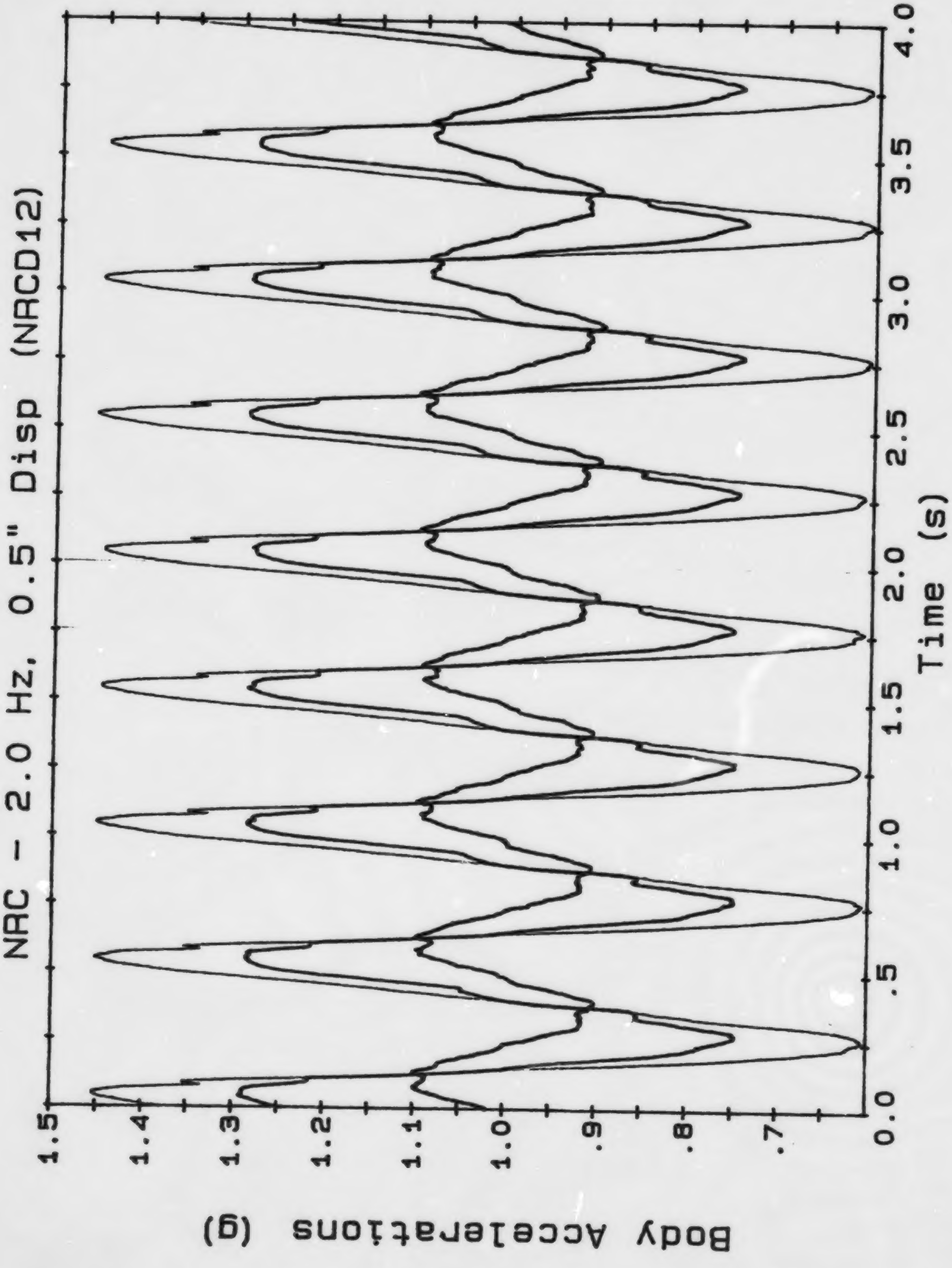

NRC - 2.0 Hz, 0.5" Disp (NRCD12)
Body Accelerations (g)
Time (s)
1.5
1.4
1.3
1.2
1.1
1.0
.9
.8
.7
0.0
.5
1.0
1.5
2.0
2.5
3.0
3.5
4.0

FIGURE 5.29

6.0 STEADY STATE HANDLING

One measure of vehicle handling performance can be determined through steady state handling tests. Steady state handling can be considered as the vehicle response to constant steer wheel input or to constant path.

6.1 TEST OBJECTIVE

The objective was to measure the steady-state handling response behaviour of the test bus.

6.2 TEST APPROACH

The test approach followed procedures outlined by the International Organization for Standardization (I.S.O.), Report 4138, entitled "Road vehicles - Steady state circular test procedures". The bus was driven around a constant radius circle at increments of increasing forward speed.

The vehicle steady-state response was measured through such variables as the road wheel angles, roll angle and slip angle of the bus. Also, such parameters as the forward velocity, lateral acceleration and handwheel angle were measured.

6.3 INSTRUMENTATION AND EQUIPMENT

The following describes the instrumentation used for steady-state handling:

Hand wheel angle	A multi-turn potentiometer was mounted under the dash and was driven by a piece of cord wrapped around the steering column. Measurement achieved ±945° ±0.2°.

Lateral Acceleration	Accelerometer mounted at vehicle C of G (vertically on the floor of the vehicle) firmly attached to a wood platform taped to the floor. Measurement achieved ±20 m/s^2 ±0.5%.
Slip Angle	Trolley mounted at rear axle on lateral centreline of vehicle Measurement achieved ±20° ±0.15°.
Roll Angle	Trolley (as for slip angle) Measurement achieved ±13° ±0.15°.
Forward velocity	Pick-up off cruise control 0-34 m/s ±0.2 m/s.
Front road wheel steer angle	Displacement transducers were mounted from centre of steering tie rod to axle ±125 mm ±1% (check stroke).
Axle steer angles	Two displacement transducers were mounted along lower axle location links at each axle (body to axle). ±10 mm ±1%.
Axle vertical	Displacement transducers were mounted vertically across dampers ±100 mm ±1%

The transducers were calibrated in the UTDC Calibration Lab.

The data acquisition equipment was essentially the same as that of the Ride Quality tests (See Section 5.3).

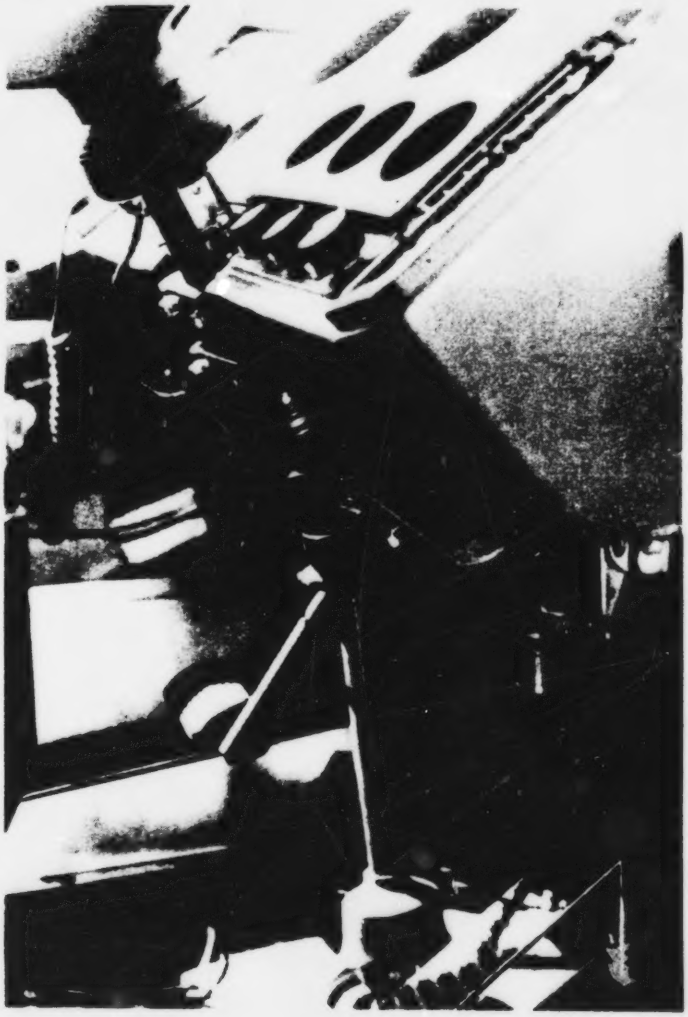


FIGURE 6.1: HANDWHEEL ANGLE TRANSDUCER

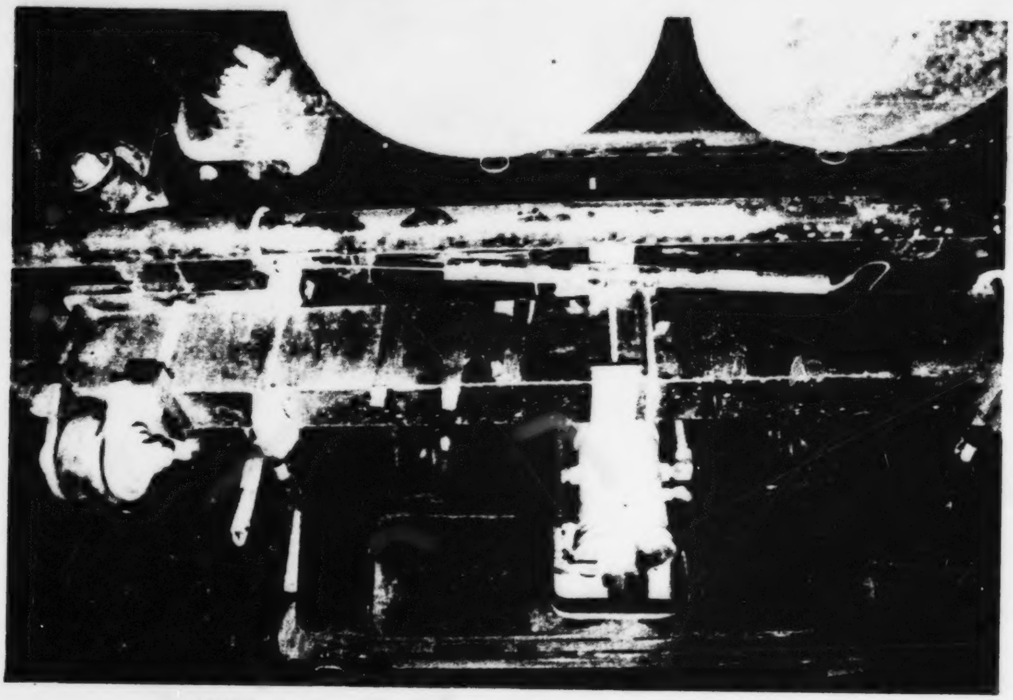


FIGURE 6.2: TIE ROD DISPLACEMENT TRANSDUCER

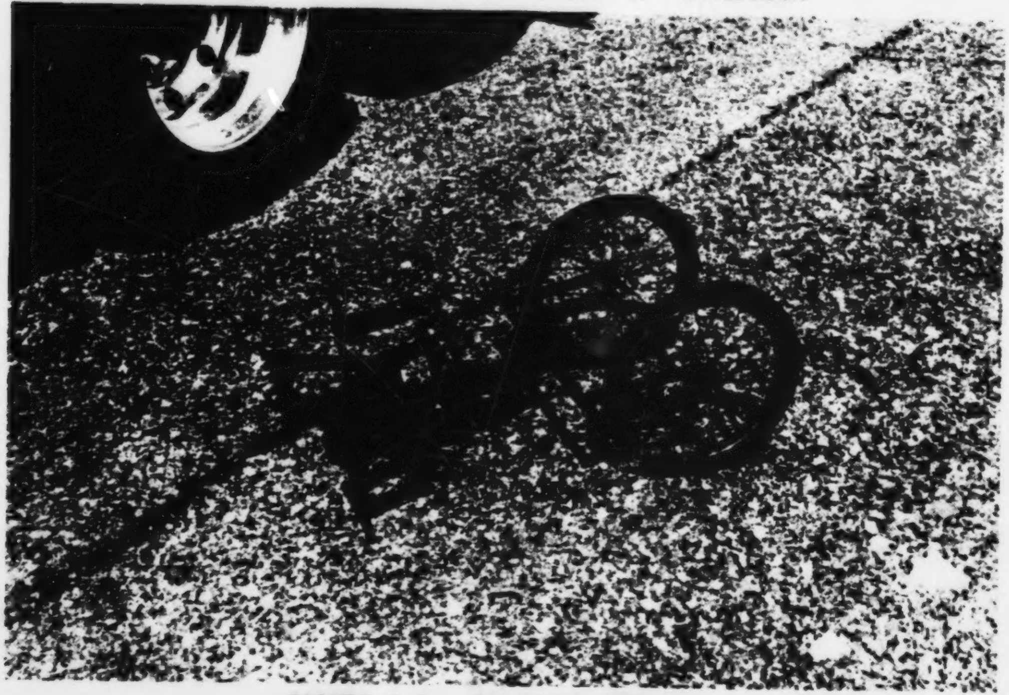


FIGURE 6.3: BALANCED TROLLEY

6.4 TEST SET-UP AND LOCATION

Tests were carried out on uniform smooth, dry, concrete and tarmac surfaces with high friction levels (tire to surface "mu" = 0.6). There was less than 2% gradient over the test areas.

The condition of the test tires, (important role in vehicle handling), was less than half worn with good shoulder condition. Care was taken that tire temperatures were not excessively high or low throughout the tests (30-45°C).

Site preparation involved painting a white circle on the paved surface. For this purpose, a mixture of lime and water was used, with a paint brush. A centre was first located, and a stretched length of nylon rope pivoted from this centre point to follow the circumference of the circle while painting.

Three separate test sites were utilized for the steady-state handling tests. The first site was in the parking lot at TTL. A 15 m radius curve was formed, and used to familiarize personnel with test techniques. The second site was at Mountain View airport at the junction of two runways where a 35 m radius circle was formed on the tarmac surface. The third site was at the Transport Canada Vehicle Test Site near Blainville, Quebec in an area of the test site known as "Zone Delta". In Zone Delta a 40 m radius circle was painted.

The test bus was set-up in the appropriate configuration before a test could begin. The fuel level was usually maintained around 3/4 full to prevent fuel spillage. The levelling valves were disconnected before the steady-state tests to ensure that the valves would not try to level the bus during the steady-state manoeuvers. Finally, instrumentation was connected as per the steady-state handling tests. This includes all instrumentation described in Section 6.3.

Tests were conducted according to the following schedule.

June 17/86	Mountainview Airport, Configuration #3.
June 19/86	Mountainview Airport, Configuration #3.
July 8/86	Blainville, Quebec, Configuration #8.
July 9/86	Blainville, Quebec, Configuration #9 and 11.

6.5 TEST PROCEDURE

1)	Vehicle Configuration	Vehicle was set to the required configuration and vehicle data was recorded on the data sheets.
2)	Tire warm up	If tires were not sufficiently warm, the bus was driven around site for about 5 minutes. If this was not possible, it was driven 500 m at a lateral acceleration of 3 m/s^2.
3)	Zero Set	Instrumentation was switched on and allowed sufficient warm up time. Next the bus was driven on the marked circle and gently coasted to a halt in a near steady state manner. Instrumentation zeros were recorded.

The vehicle was driven at the lowest possible speed on the circle. With the steering wheel and throttle position fixed, at least 3 seconds of data were recorded using the highest gear compatible with a smooth driving technique at each test condition. Lateral acceleration was increased in 0.2 m/s^2 increments taking at least 3 seconds of data at each point with the steering and throttle held fixed by the driver. The path did not vary more than 0.3 m either side of the painted circle. The lateral acceleration was increased until the driver no longer felt able to hold a steady state condition (approximately 4.3 - 4.7 m/s^2). Finally, the bus was coasted to zero velocity, keeping on the circle. When the bus reached zero forward speed, all instrumentation zeros were again recorded.

The test was carried out for left and right hand turns.

6.6 TEST RESULTS

Steady State Handling results are shown in figures 6.4 and 6.5. Yaw angle of the balanced slip trolley and roll angle are fit (using a second order fit) to lateral body acceleration at the c.g.

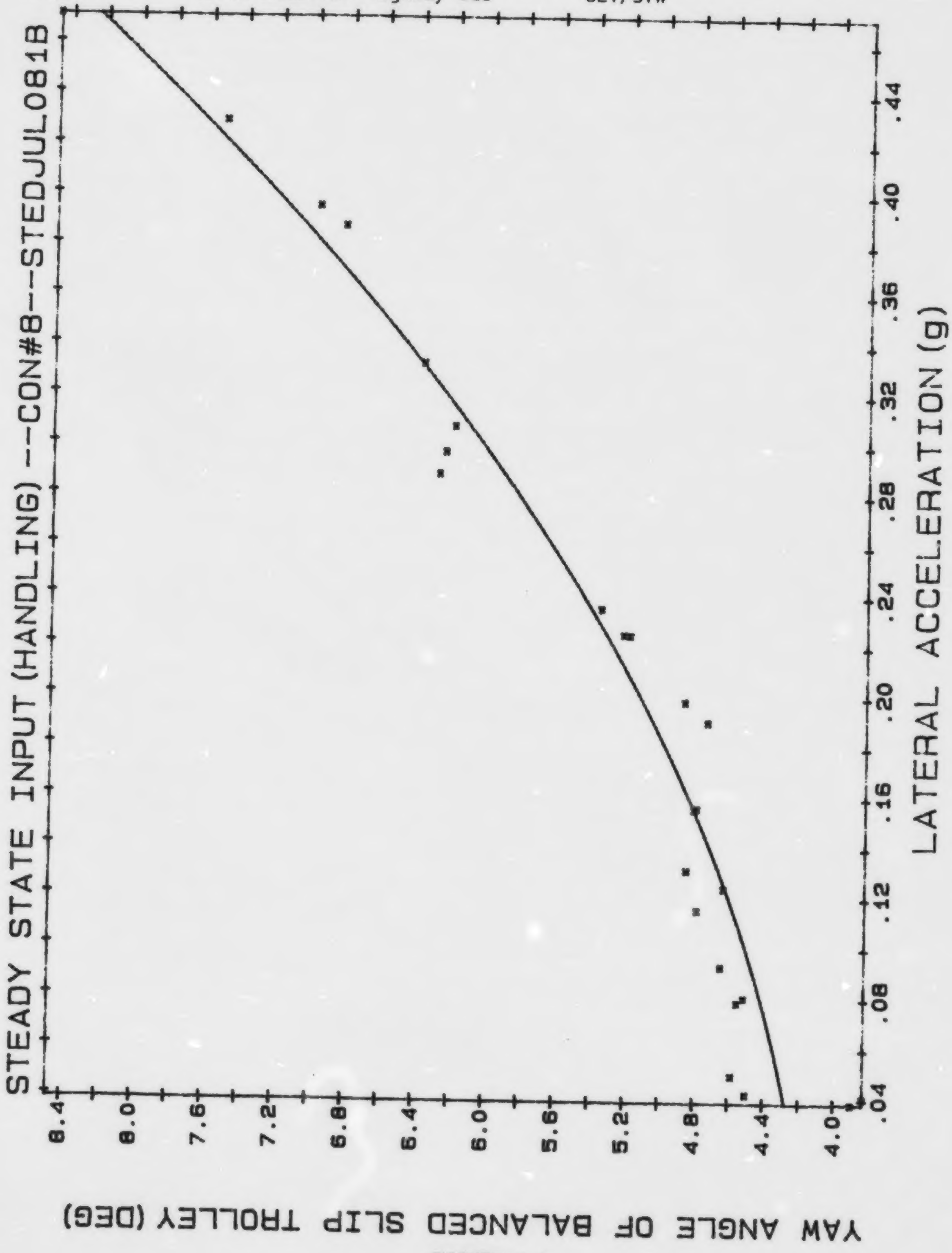

STEADY STATE INPUT (HANDLING) --CON#8--STEDJUL081B
8.4
8.0
7.6
7.2
6.8
6.4
6.0
5.6
5.2
4.8
4.4
4.0
YAW ANGLE OF BALANCED SLIP TROLLEY (DEG)
.04
.08
.12
.16
.20
.24
.28
.32
.36
.40
.44
LATERAL ACCELERATION (g)

FIGURE 6.4

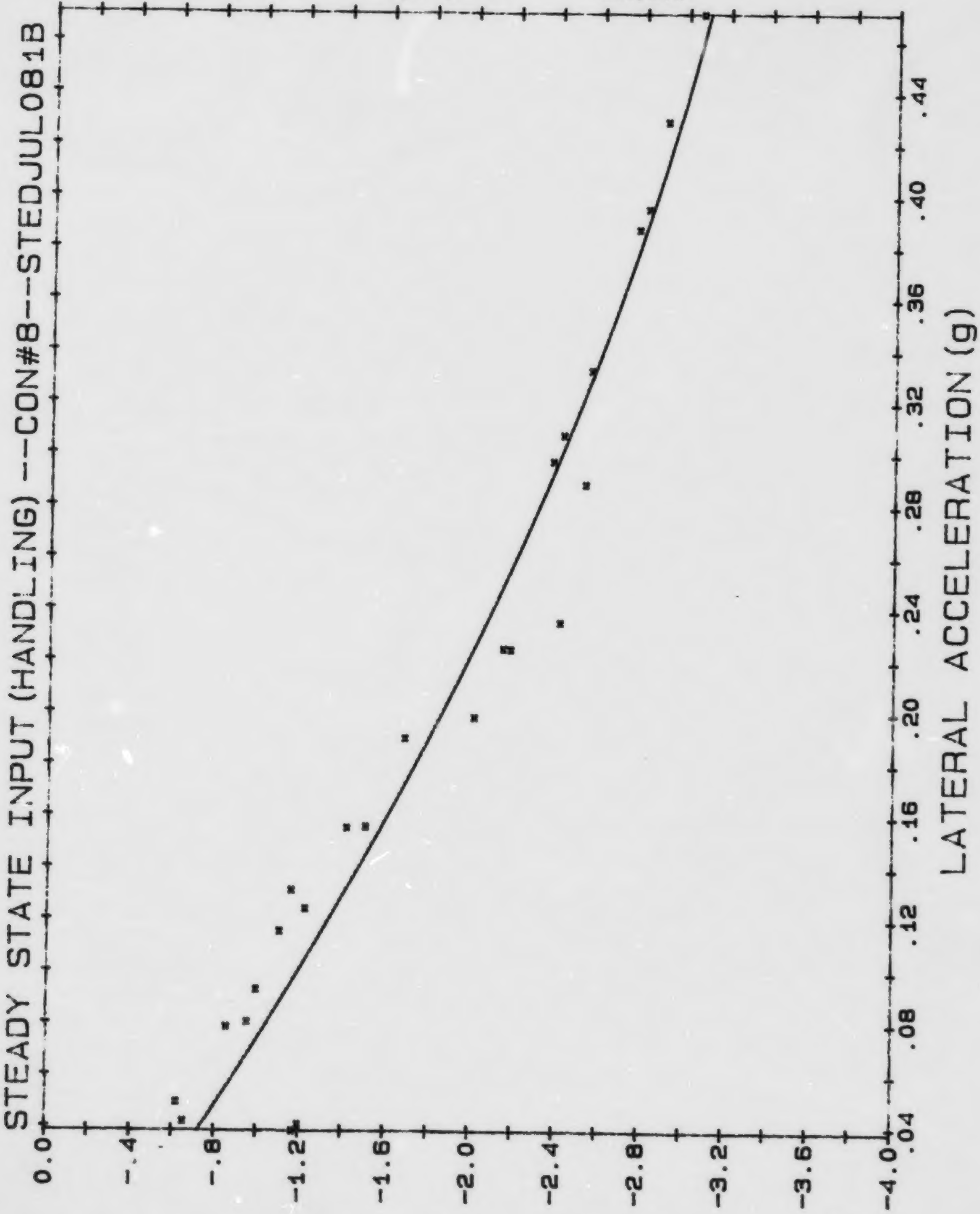

STEADY STATE INPUT (HANDLING) --CON#8--STEDJUL081B
LATERAL ACCELERATION (g)
.04
.08
.12
.16
.20
.24
.28
.32
.36
.40
.44
ROLL ANGLE (DEG)
0.0
-.4
-.8
-1.2
-1.6
-2.0
-2.4
-2.8
-3.2
-3.6
-4.0

FIGURE 6.5

7.0 TRANSIENT HANDLING

Another measure of vehicle handling performance is through tests of transient handling response.

The chosen tests followed procedures outlined by the International Organization for Standardization (I.S.O.), Draft Proposal 7401, "Road Vehicles - Lateral Transient Response Test Procedures", September 1984.

7.1 TEST OBJECTIVES

The primary objective is to measure the transient handling response behaviour of the test bus in both the time and frequency domains.

7.2 TEST APPROACH

Transient handling test results can be divided into two categories: Those involving a time response and those involving a frequency response. For this test program, one test input was chosen from each of the two categories. From the time response category, a Step Input was selected. From the frequency response category, Random Input was selected.

7.2.1 Time Domain-Step Input

For the case of a step input, the test bus was driven at constant speed in a straight line, and at a selected point, a sudden steering input was induced causing the bus to turn sharply while maintaining throttle. This pseudo-step (ramp) input was held constant for several seconds allowing all transients to decay. A time history was recorded of the step input and the response of the test bus.

7.2.2 Frequency Domain-Random Input

The Random Input test began with the bus driven at constant speed in a straight line. At some point, the driver began steering "erratically" by moving the handwheel clockwise and anti-clockwise in a sinusoidal motion. Initially the frequency low (0.2 Hz) but was gradually increased to higher frequencies (1.5 Hz) trying to keep the handwheel turn amplitude approximately constant (±90°). During the run, the speed is held constant. When sufficient frequencies had been covered, or the driving space was becoming too short, the bus was again driven straight at constant speed to complete the test.

7.3 INSTRUMENTATION AND EQUIPMENT

The following describes the Instrumentation for Transient Handling:

Handwheel Angle	A multi-turn potential was mounted under the dash and was driven by a piece of cord wrapped around the steering column. Measurement achieved ±945° ±0.2°.
Lateral Acceleration	Accelerometer mounted at vehicle C of G (vertically on the floor of the vehicle) firmly attached to a wood platform taped to the floor. Measurement achieved ±20 m/s^2 ±0.5%.

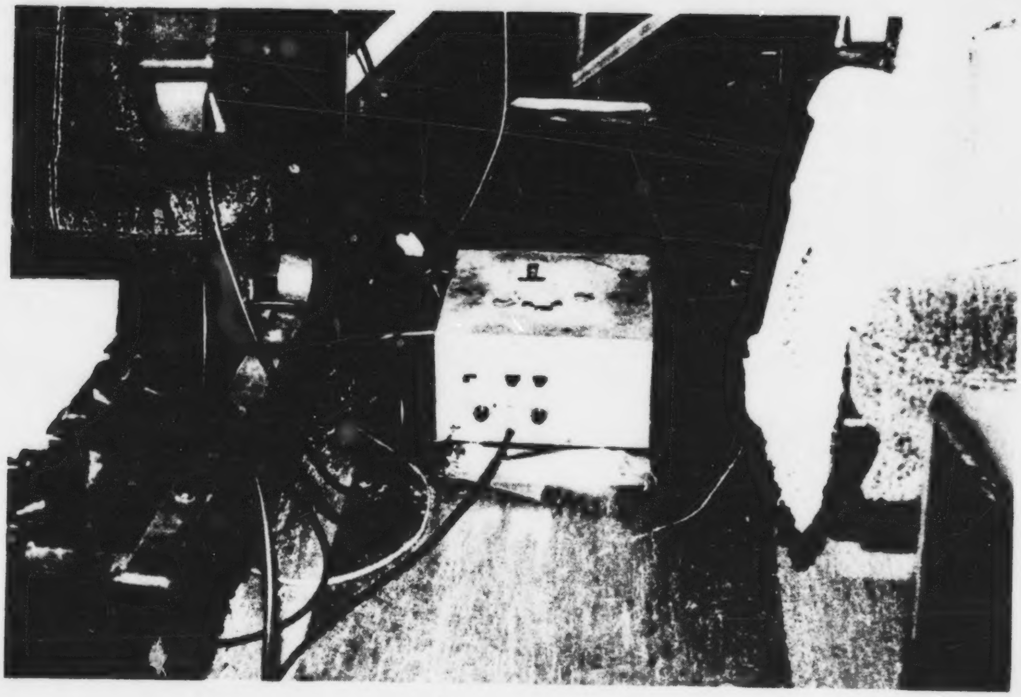


FIGURE 7.1: RATE GYRO

Yaw velocity	Rate gyro Measurement achieved 50°/s ±0.5°/s
Forward velocity	Pick-up off cruise control 0-34 m/s ±0.2 m/s
Front road wheel steer angle	Displacement transducers were mounted from centre of steering tie rod to axle ±125 mm ±1% (check stroke) (mean road wheel angle required)
Axle steer angles	Two displacement transducers were mounted longitudinally between the body and front axle ±10 mm ±1%
Axle vertical	Displacement transducers were mounted vertically across dampers ±100 mm ±1%

The transducers were calibrated in the TTL workshop.

The data acquisition equipment was essentially the same as that of the Ride Quality Tests (See Section 5.3).

7.4 TEST SET-UP AND LOCATION

Tests were carried out on uniform, smooth, dry concrete and tarmac surfaces with high friction levels (tire surface "mu" greater than 0.6). There was less than 2% gradient over the test areas.

The condition of the tires (important in vehicle handling) was less than half worn with good shoulder condition. Care was taken that their temperatures were not excessively high or low throughout the tests (30-45°C).

No site preparation was required as with the steady-state handling tests. However, before tests could be started it was ensured that there was adequate space to perform a test. This was usually accomplished by driving through a "mock" test.

Several test sites were used for performing transient handling tests. While practicing step inputs, the TTL parking lot was used. For practicing random inputs, Taylor-Kidd Boulevard near TTL was used. Actual transient handling tests were performed at Mountainview Airport (Belleville) and the Transport Canada Vehicle test site near Blainville, Quebec. At Blainville, step input handling tests were conducted in the dynamic test area known as "Zone Delta". Random input tests were conducted on the high speed test track known as "Zone Bravo".

Tests were conducted according to the following schedule:

June 17/86	Mountainview Airport, Configuration #3. (Step and Transient)
June 19/86	Mountainview Airport, Configuration #3, 4 (Step and Transient)
July 8/86	Blainville, Quebec, Configuration #8, (Step and Transient)
July 9/86	Blainville, Quebec, Configuration #9, 10, and 11. (Step and Transient)

7.5 TEST PROCEDURE

1)	Vehicle Configuration	Vehicle was set to the required test configuration and vehicle data was recorded on the data sheets.
2)	Tire warm up	If the tires were not already warm, then the tires were warmed by driving at about 80 km/hr for 10 km. If this was not possible, the bus was driven 500 m at a lateral acceleration of 3 m/s^2.
3)	Zero Set	All instrumentation was switched on and allowed sufficient warm up time. The bus was driven in a straight line at slow speed for 10 seconds and instrumentation zeros were recorded.

7.5.1 Random Input

Speed	- Constant 80 km/hr
Max. Handwheel Angle	- To give steady state lateral acceleration of 2 m/s^2. (approx. ±90°).

The test area was approached at constant 80 km/hr in a straight line.

Data recording started 5 seconds of zero input. Following this, handwheel input of the desired angle (approx. ±90°) in sinusoidal form at low frequency (less than 0.2 Hz) was begun.

The frequency was gradually increased to include all frequencies up to 2 Hz, even if this meant the angle was reduced for higher frequencies.

Finally, 5 seconds of zero input were recorded.

Total input section was approximately of 30 seconds. Tests were shortened or lengthened to accommodate the available space.

During the tests, the amplitude PSD of Handwheel Angle was examined with the on-board micro. An attempt was made to "fill-in" the curve at all frequencies with sufficient amplitude. This meant communicating with the driver which frequencies he was presently steering at and which frequencies were required. Generally, 15 runs with 30 seconds of data were required to complete the test.

7.5.2 Step Input

Speed - Constant 40 km/hr

Handwheel angle - To give steady state lateral acceleration = 3.2 m/s^2 (approximately 225°).

The vehicle was driven in a straight line at 50 km/hr and data recording commenced. The steering input was applied as rapidly as possible (300-400°/s) (to 225°) for 3.2 m/s^2 lateral acceleration and held for several seconds so that all measured variables reached a steady state. No change of throttle was made during the manoeuvre.

Each test was usually performed three times.

Both left and right hand turns were made.

7.6 METHOD OF ANALYSIS

7.6.1 Random Input

Before analysis, each channel of data should be filtered to remove all signals higher than 15 Hz.

a) Digitize data at 40 steps/second.

b) Average left and right road wheel angles to produce a mean value.

c) Integrate roll and yaw rates to give roll angle and yaw angle data (low frequency cut-off 0.1 Hz).

d) Correct lateral acceleration data for roll angle.

e) Analyse correlation between input (handwheel angle and mean road wheel angle) and output (yaw rate, lateral acceleration, roll rate, roll angle, yaw angle).

f) F.F.T. (Fast Fourier Transformer) analysis of each variable gives power spectral density values vs frequency (Narrow band analysis with a minimum of 250 lines of equal band width from 0-5 Hz. (gives 0.02 Hz/band).) This is principally of interest to check that the content of the input is adequate. The criterion of adequacy is that broadly the range of input amplitude should not exceed 18 db (8:1) over the frequency range of interest (0.2 Hz - 1.2 Hz).

g) From the above analyse the gain of the output/input. The graphs of particular interest are yaw rate gain lateral acceleration gain and, to a lesser extent, roll angle gain plotted against frequency for both handwheel input and road wheel input.

h) Plot phase between input (handwheel angle) and output (yaw rate) lateral acceleration roll angle etc.

i) Plot time delay between input and output.

j) Curve comparison techniques with a 2 or 3 degree of freedom model (yaw, lateral and roll) of the gain and phase results can be used to estimate the natural frequencies and damping coefficients of the vehicle in these modes.

7.6.2 Step Input

a) The time histories should be individually analysed. (Note: It is likely that there will be no overshoot of yaw velocity. Leave blank any values that cannot be measured accurately.)

b) Average left hand turns and right hand turns separately. Finally average these.

7.7 RESULTS

Some typical time wave forms are shown in figures 7.2 - 7.17. Wheel angle, lateral acceleration, yaw rate and roll rate are plotted versus time.

Figure 7.2 - 7.9 present step input transient handling plots. Figure 7.10 - 7.17 present random steer transient handling plots.

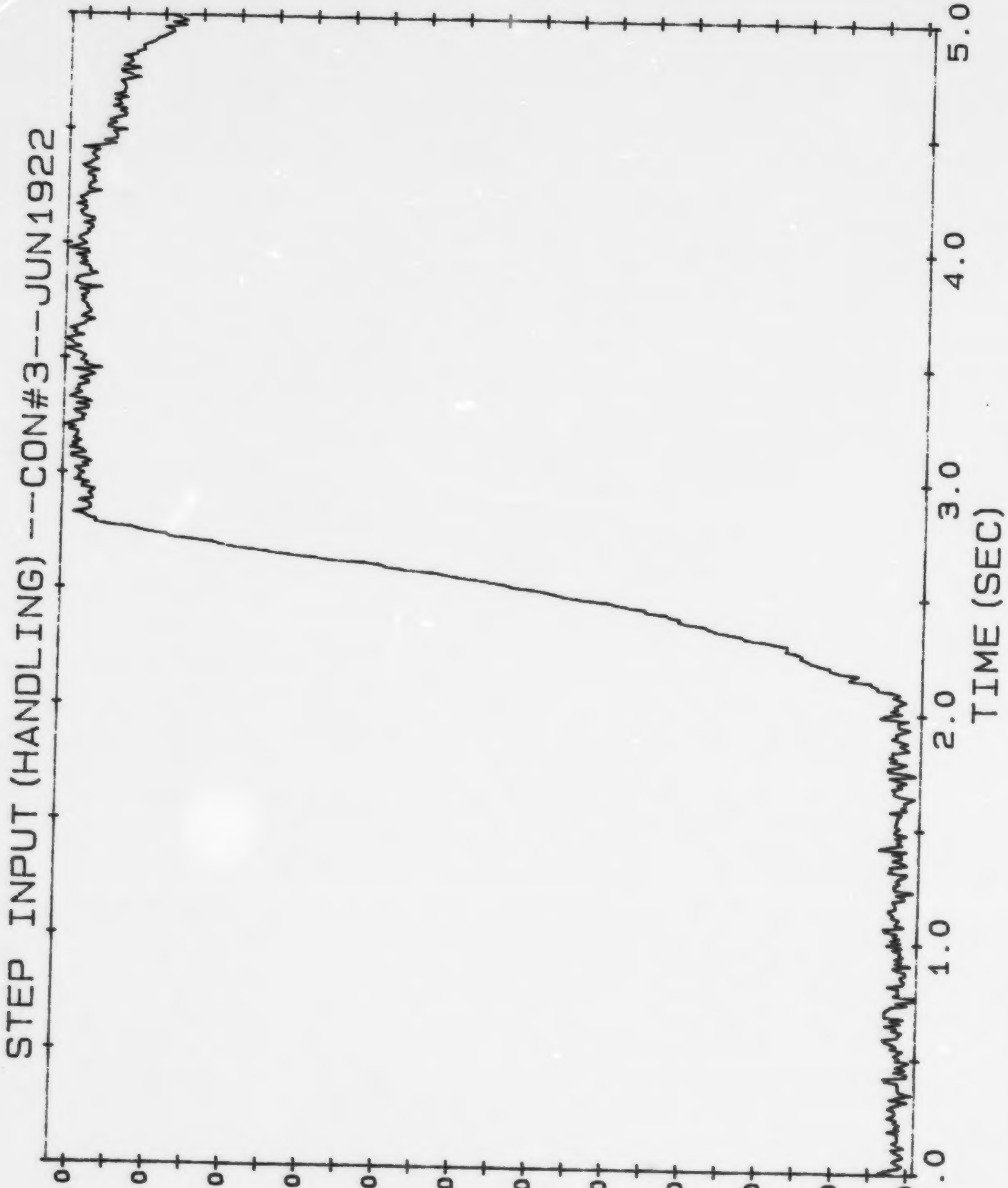

STEP INPUT (HANDLING) --CON#3--JUN1922
TIME (SEC)
0.0
1.0
2.0
3.0
4.0
5.0
WHEEL ANGLE (DEG)
1.0
2.0
3.0
4.0
5.0
6.0
7.0
8.0
9.0
10.0
11.0
12.0

FIGURE 7.2

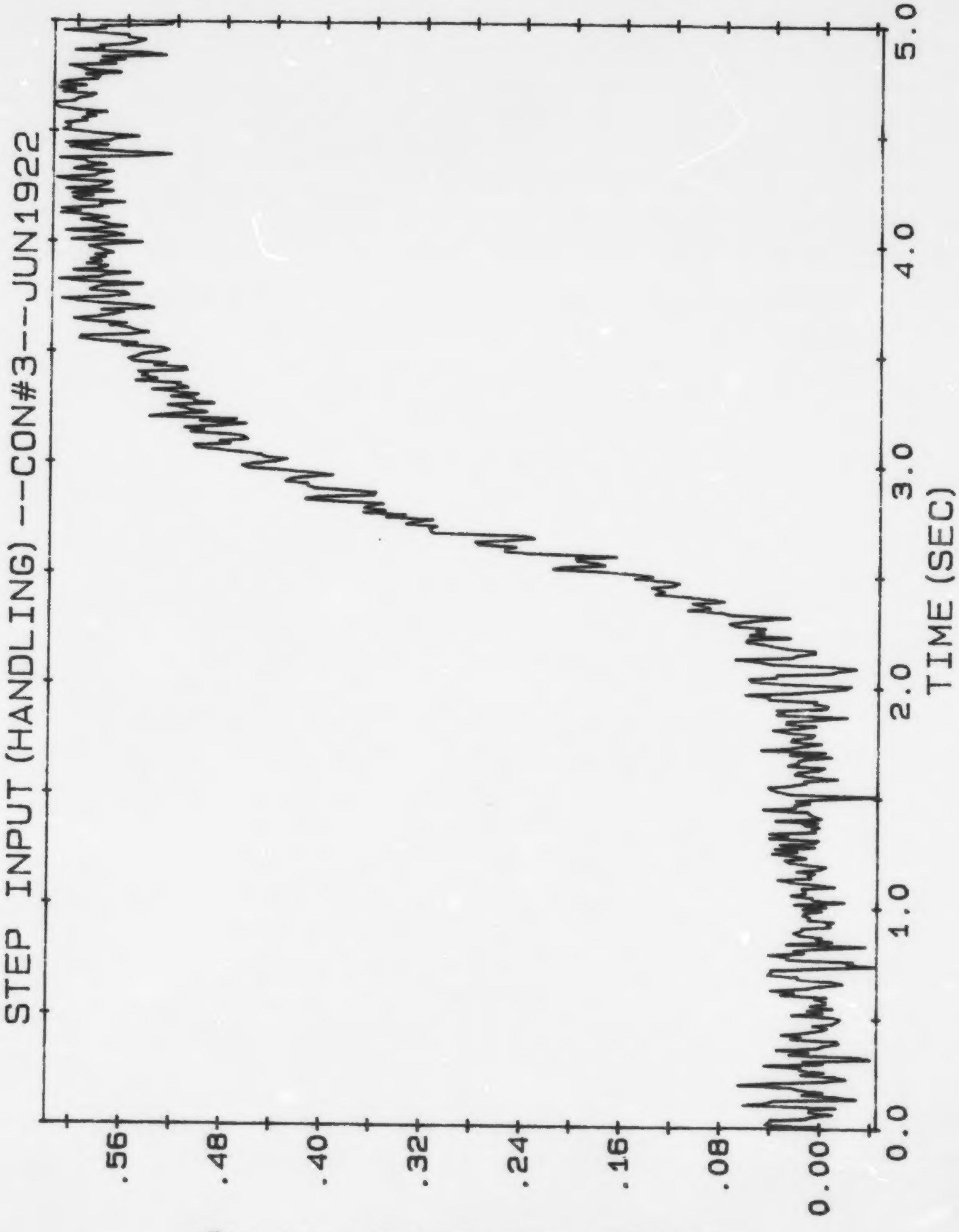

STEP INPUT (HANDLING) --CON#3--JUN1922
LATERAL ACCELERATION (g)
.56
.48
.40
.32
.24
.16
.08
0.00
TIME (SEC)
0.0
1.0
2.0
3.0
4.0
5.0

FIGURE 7.3

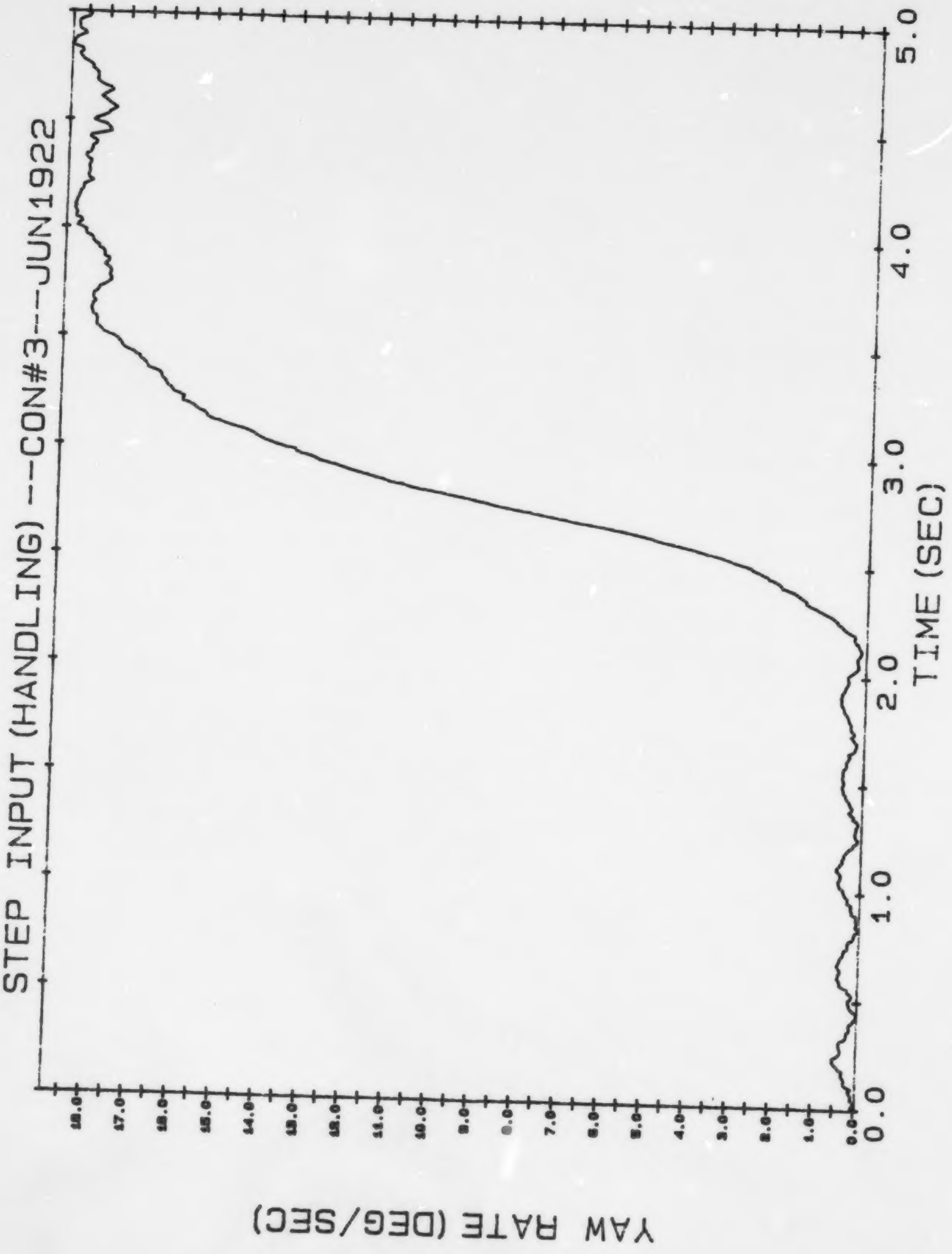

STEP INPUT (HANDLING) --CON#3--JUN1922
TIME (SEC)
0.0
1.0
2.0
3.0
4.0
5.0
YAW RATE (DEG/SEC)
0.0
1.0
2.0
3.0
4.0
5.0
6.0
7.0
8.0
9.0
10.0
11.0
12.0
13.0
14.0
15.0
16.0
17.0
18.0

FIGURE 7.4

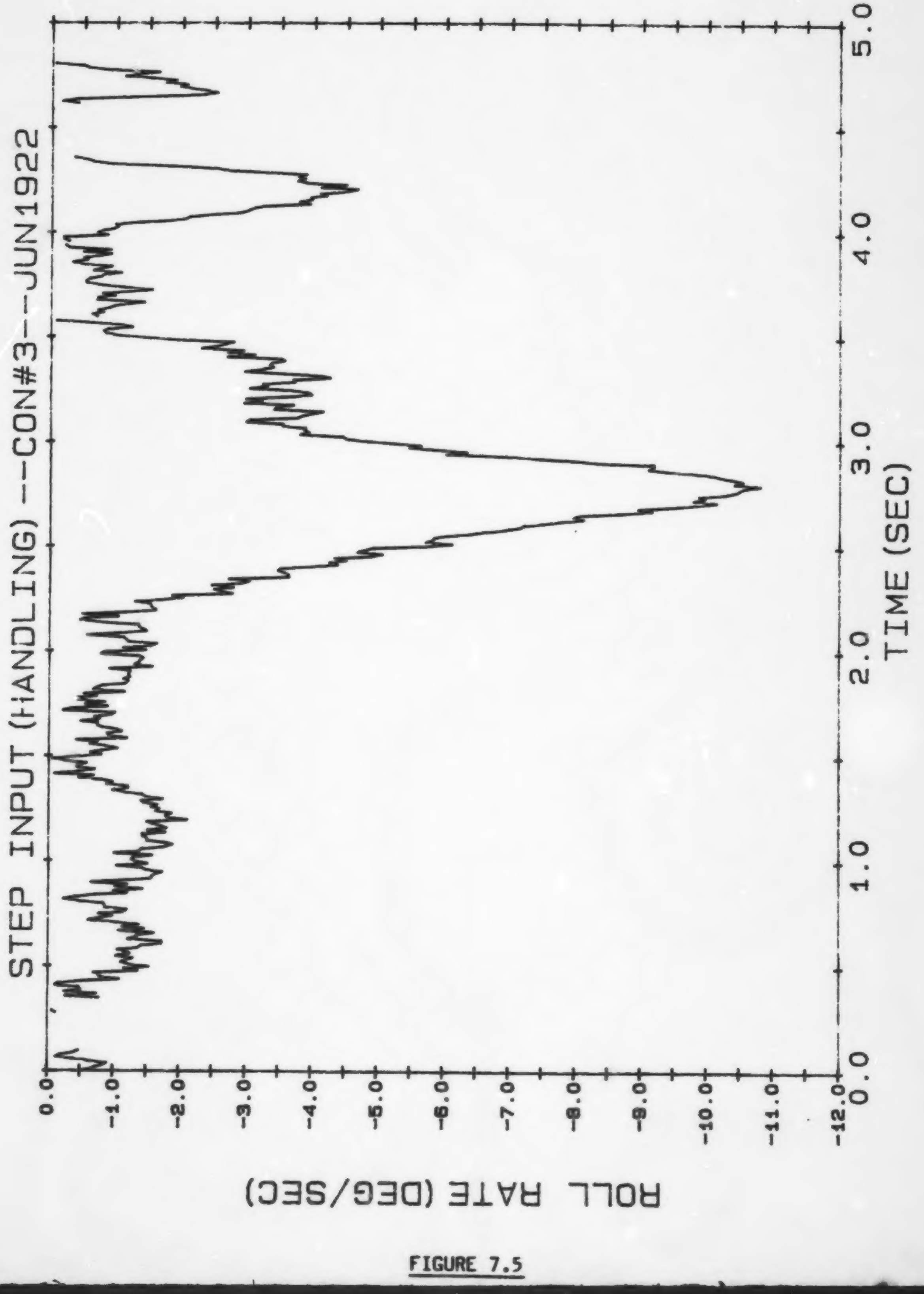

STEP INPUT (HANDLING) --CON#3--JUN1922
ROLL RATE (DEG/SEC)
TIME (SEC)
0.0
-1.0
-2.0
-3.0
-4.0
-5.0
-6.0
-7.0
-8.0
-9.0
-10.0
-11.0
-12.0
0.0
1.0
2.0
3.0
4.0
5.0

FIGURE 7.5

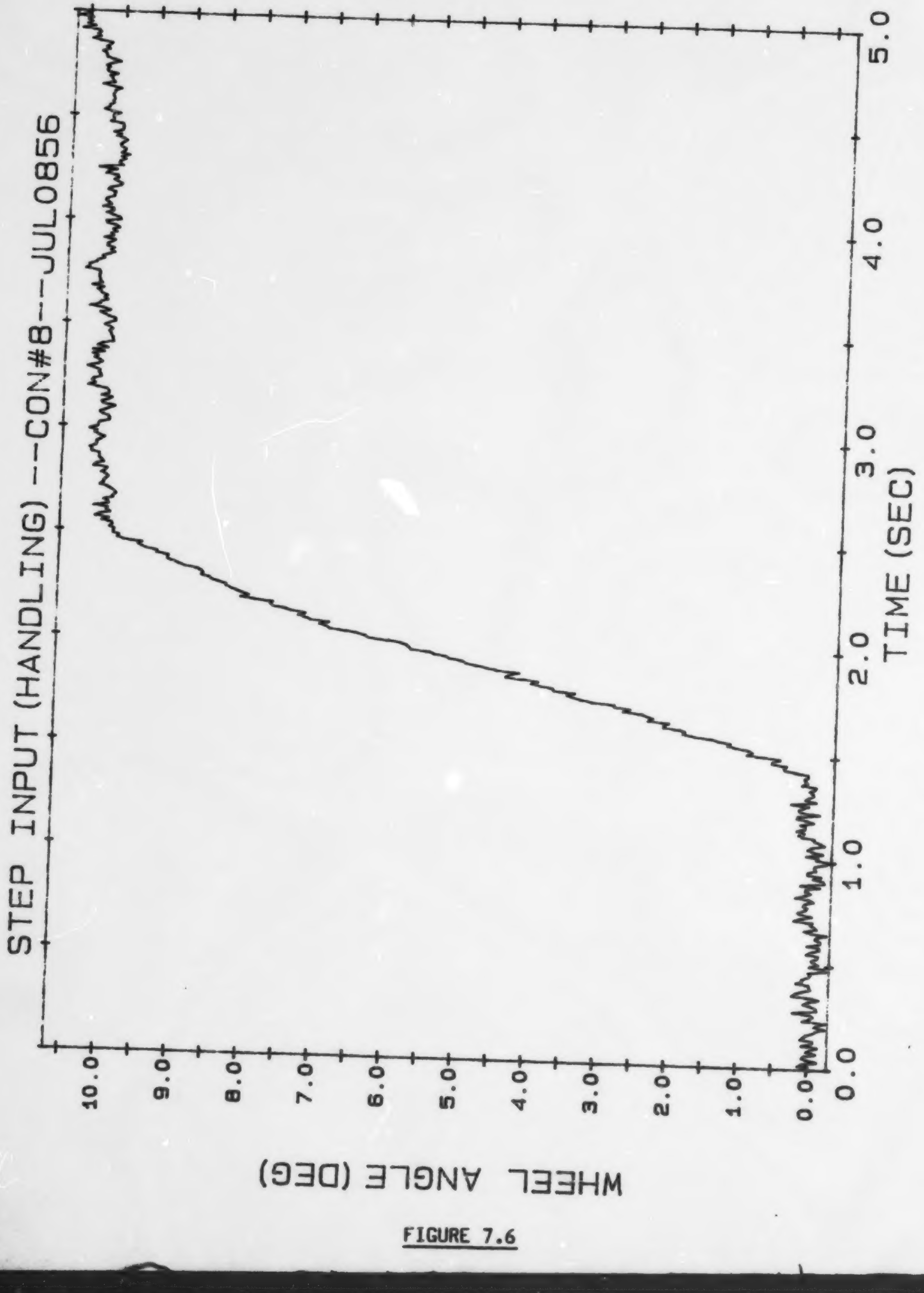

STEP INPUT (HANDLING) --CON#8--JUL0856
WHEEL ANGLE (DEG)
10.0
9.0
8.0
7.0
6.0
5.0
4.0
3.0
2.0
1.0
0.0
TIME (SEC)
0.0
1.0
2.0
3.0
4.0
5.0

FIGURE 7.6

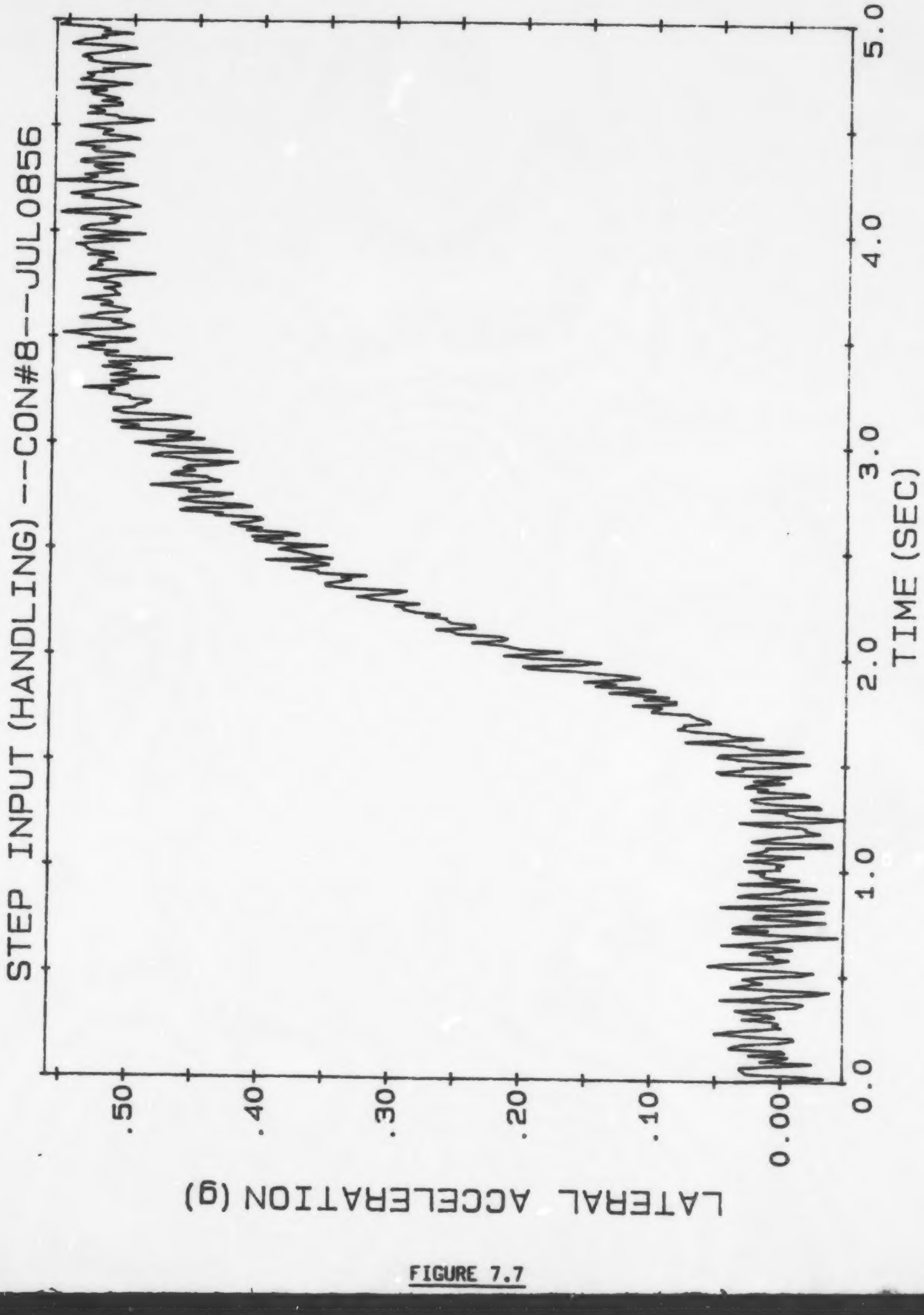

STEP INPUT (HANDLING) --CON#8--JUL0856
TIME (SEC)
0.0
1.0
2.0
3.0
4.0
5.0
LATERAL ACCELERATION (g)
0.00
.10
.20
.30
.40
.50

FIGURE 7.7

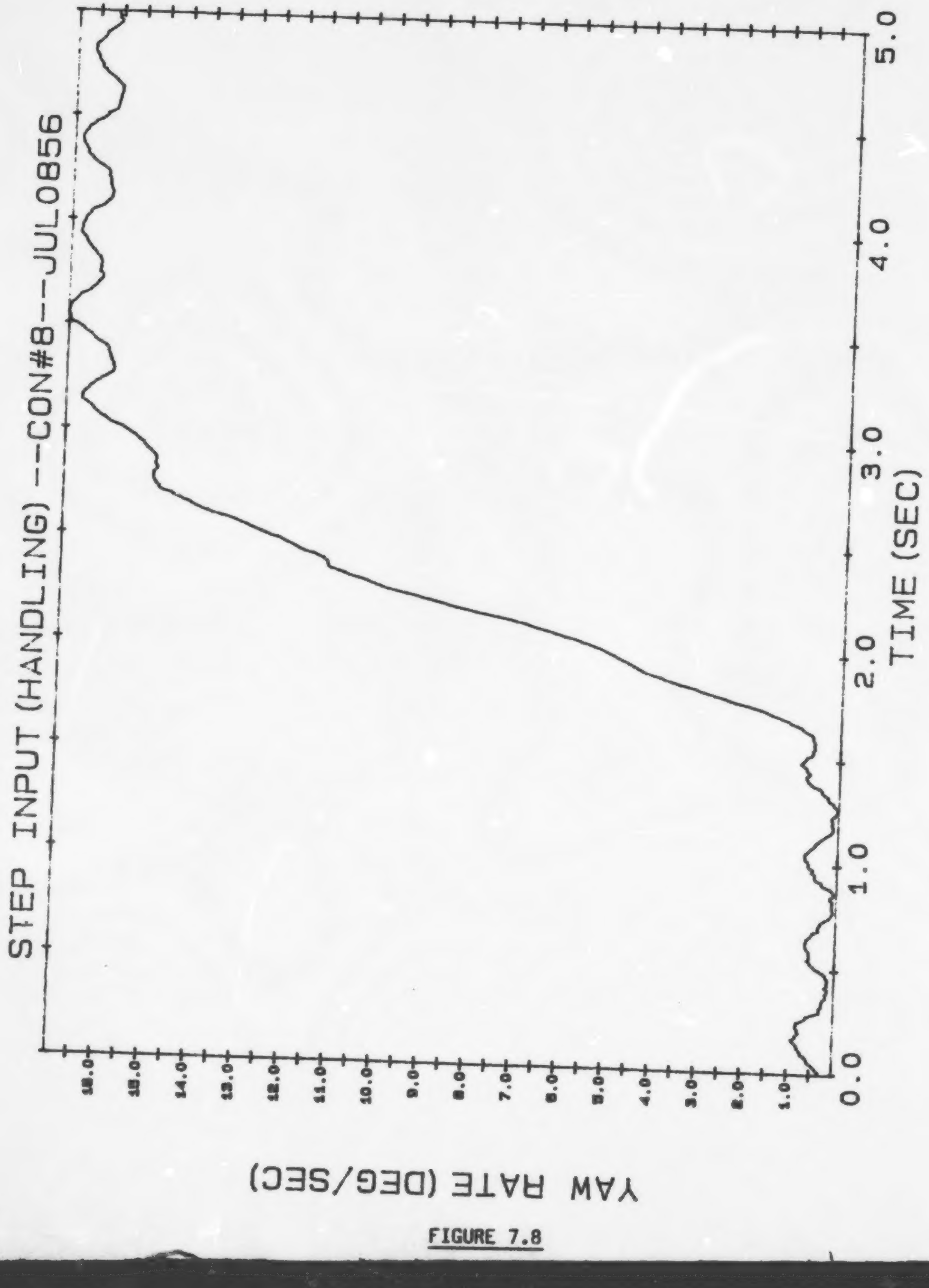

STEP INPUT (HANDLING) --CON#8--JUL0856
TIME (SEC)
0.0
1.0
2.0
3.0
4.0
5.0
YAW RATE (DEG/SEC)
1.0
2.0
3.0
4.0
5.0
6.0
7.0
8.0
9.0
10.0
11.0
12.0
13.0
14.0
15.0
16.0

FIGURE 7.8

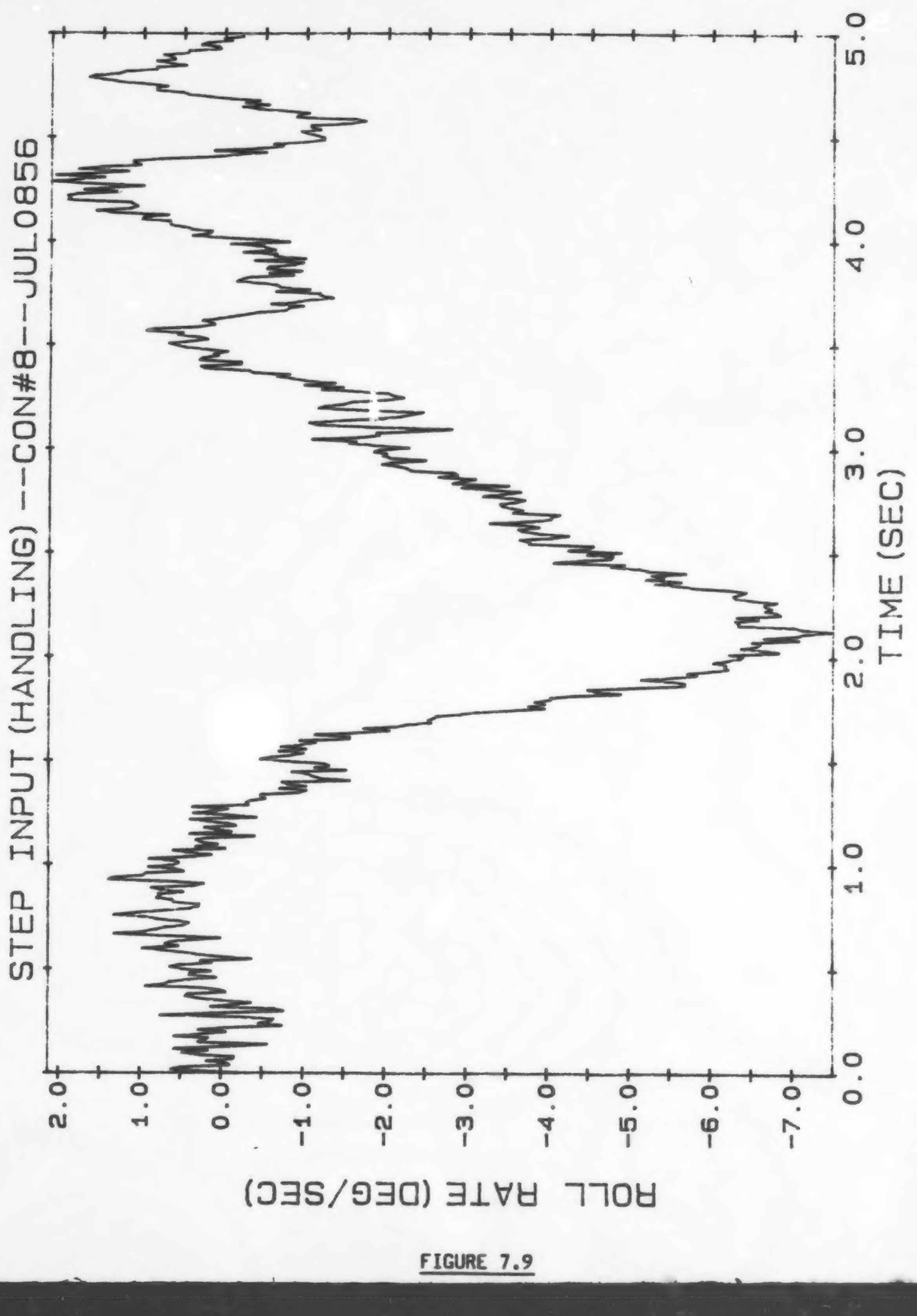

STEP INPUT (HANDLING) --CON#8--JUL0856
ROLL RATE (DEG/SEC)
TIME (SEC)
2.0
1.0
0.0
-1.0
-2.0
-3.0
-4.0
-5.0
-6.0
-7.0
0.0
1.0
2.0
3.0
4.0
5.0

FIGURE 7.9

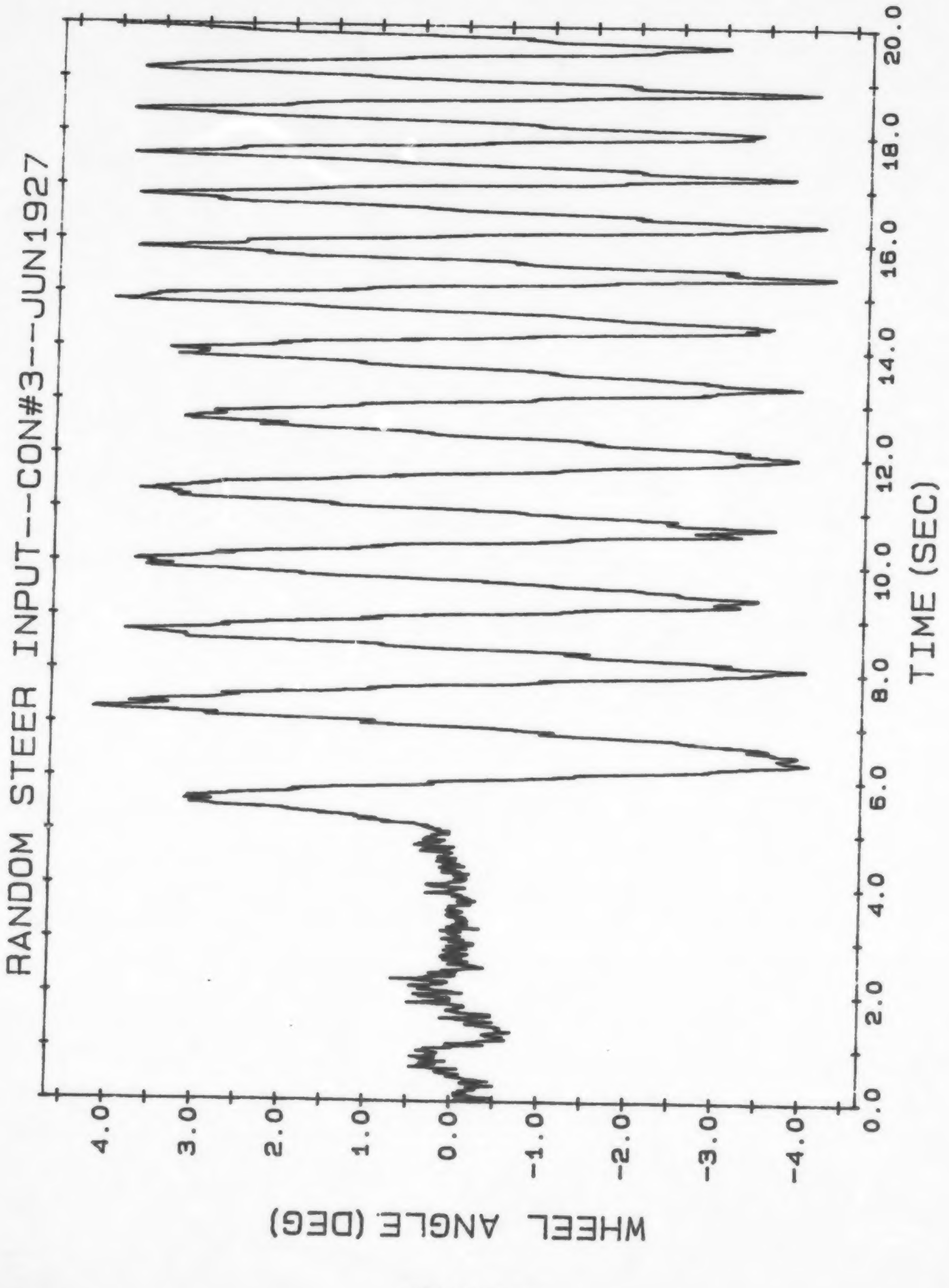

RANDOM STEER INPUT--CON#3--JUN1927
TIME (SEC)
0.0
2.0
4.0
6.0
8.0
10.0
12.0
14.0
16.0
18.0
20.0
WHEEL ANGLE (DEG)
4.0
3.0
2.0
1.0
0.0
-1.0
-2.0
-3.0
-4.0

FIGURE 7.10

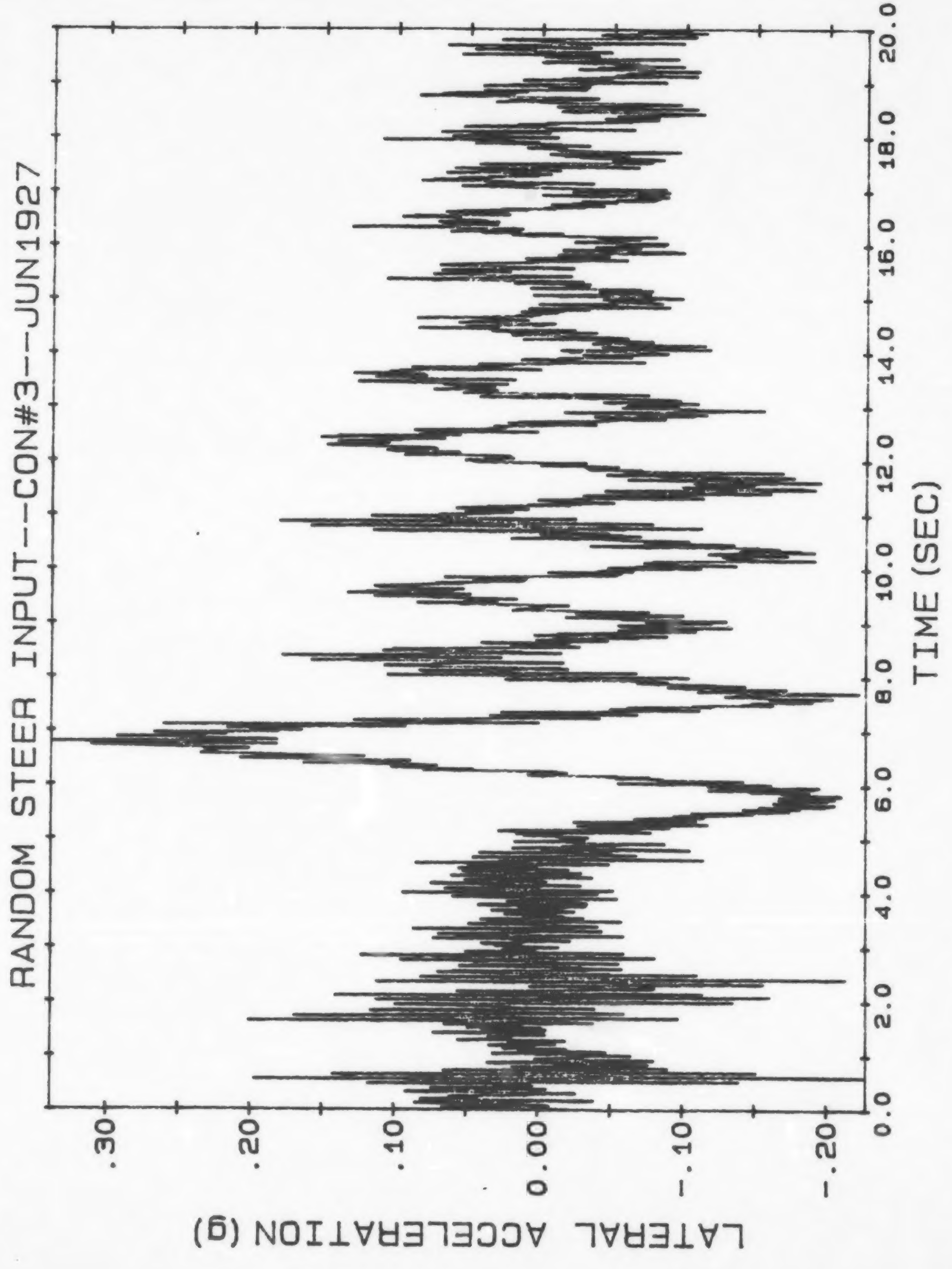

RANDOM STEER INPUT--CON#3--JUN1927
TIME (SEC)
0.0
2.0
4.0
6.0
8.0
10.0
12.0
14.0
16.0
18.0
20.0
LATERAL ACCELERATION (g)
.30
.20
.10
0.00
-.10
-.20

FIGURE 7.11

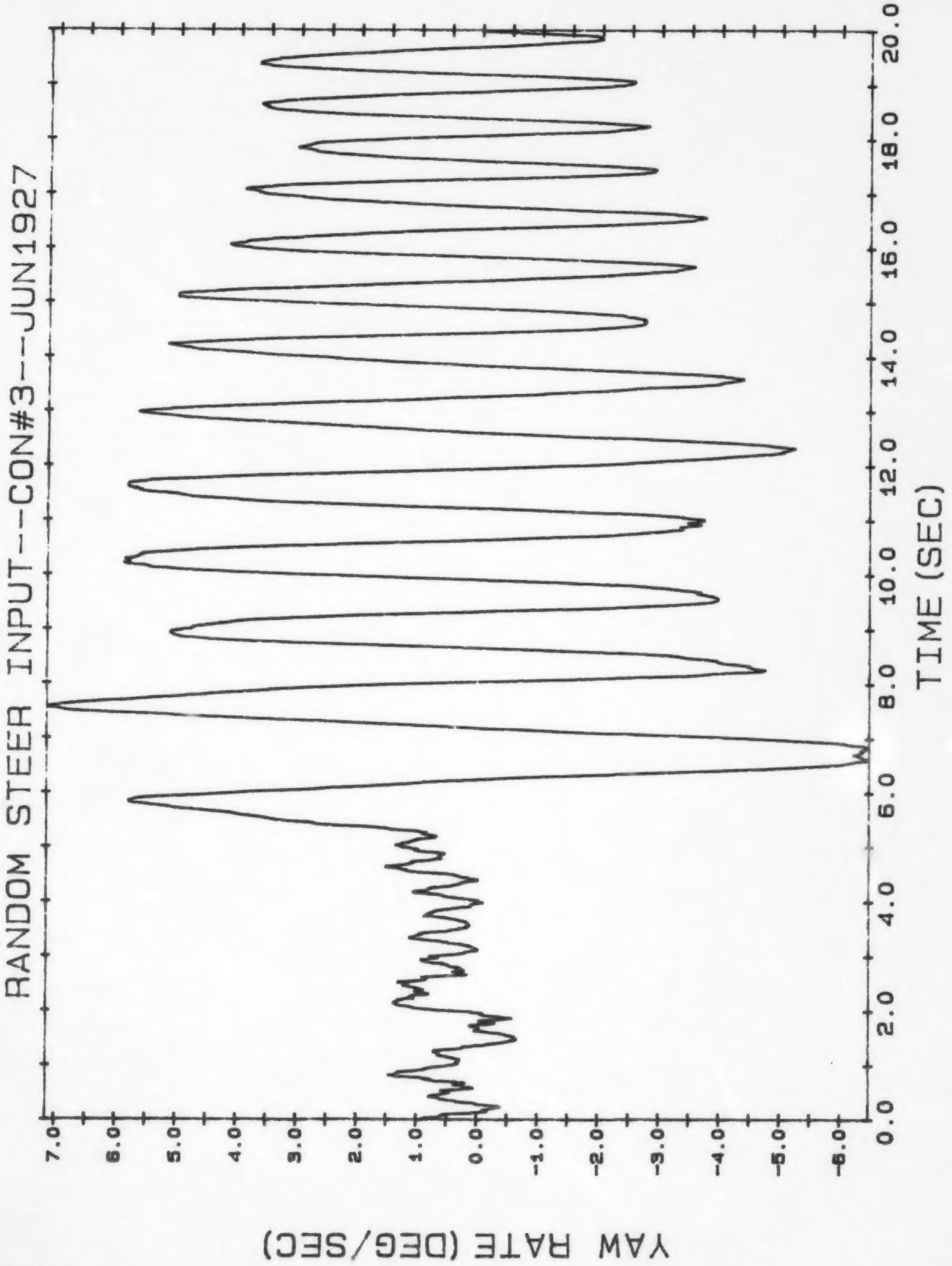

RANDOM STEER INPUT--CON#3--JUN1927
TIME (SEC)
0.0
2.0
4.0
6.0
8.0
10.0
12.0
14.0
16.0
18.0
20.0
YAW RATE (DEG/SEC)
7.0
6.0
5.0
4.0
3.0
2.0
1.0
0.0
-1.0
-2.0
-3.0
-4.0
-5.0
-6.0

FIGURE 7.12

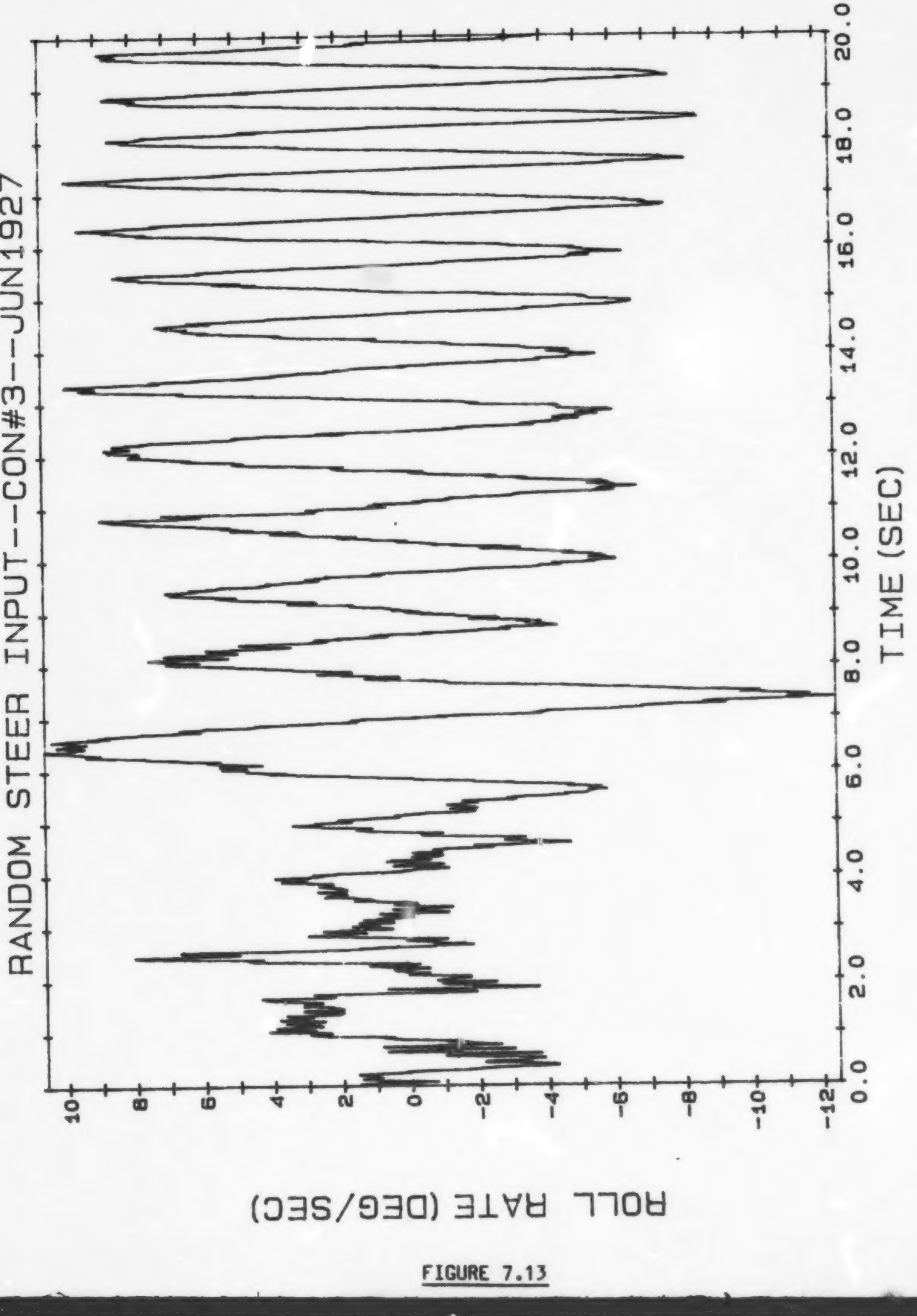

RANDOM STEER INPUT--CON#3--JUN1927
TIME (SEC)
0.0
2.0
4.0
6.0
8.0
10.0
12.0
14.0
16.0
18.0
20.0
ROLL RATE (DEG/SEC)
10
8
6
4
2
0
-2
-4
-6
-8
-10
-12

FIGURE 7.13

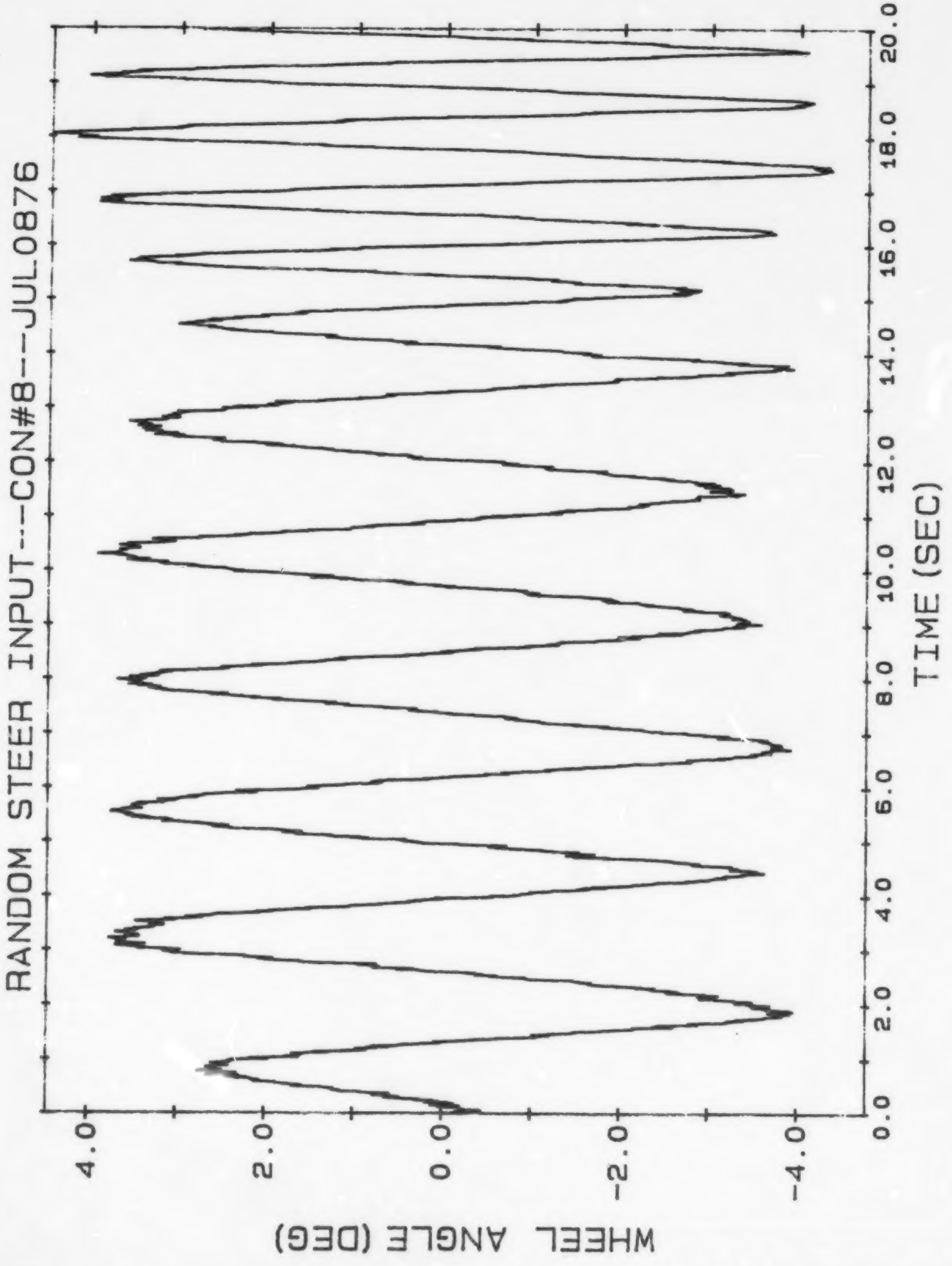

RANDOM STEER INPUT---CON#8---JUL0876
TIME (SEC)
0.0
2.0
4.0
6.0
8.0
10.0
12.0
14.0
16.0
18.0
20.0
WHEEL ANGLE (DEG)
4.0
2.0
0.0
-2.0
-4.0

FIGURE 7.14

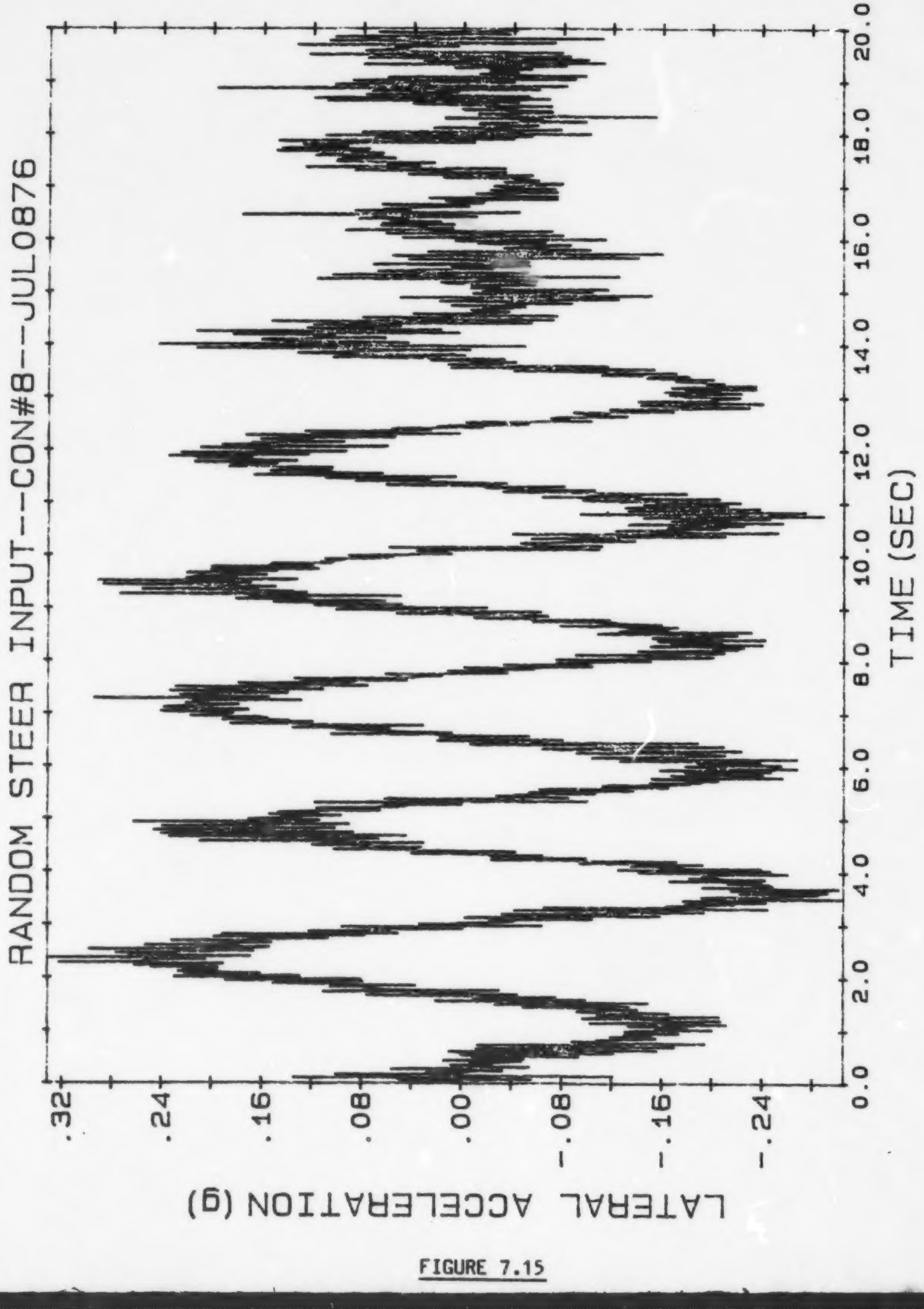

RANDOM STEER INPUT--CON#8--JUL0876
0.0
2.0
4.0
6.0
8.0
10.0
12.0
14.0
16.0
18.0
20.0
TIME (SEC)
.32
.24
.16
.08
.00
-.08
-.16
-.24
LATERAL ACCELERATION (g)

FIGURE 7.15

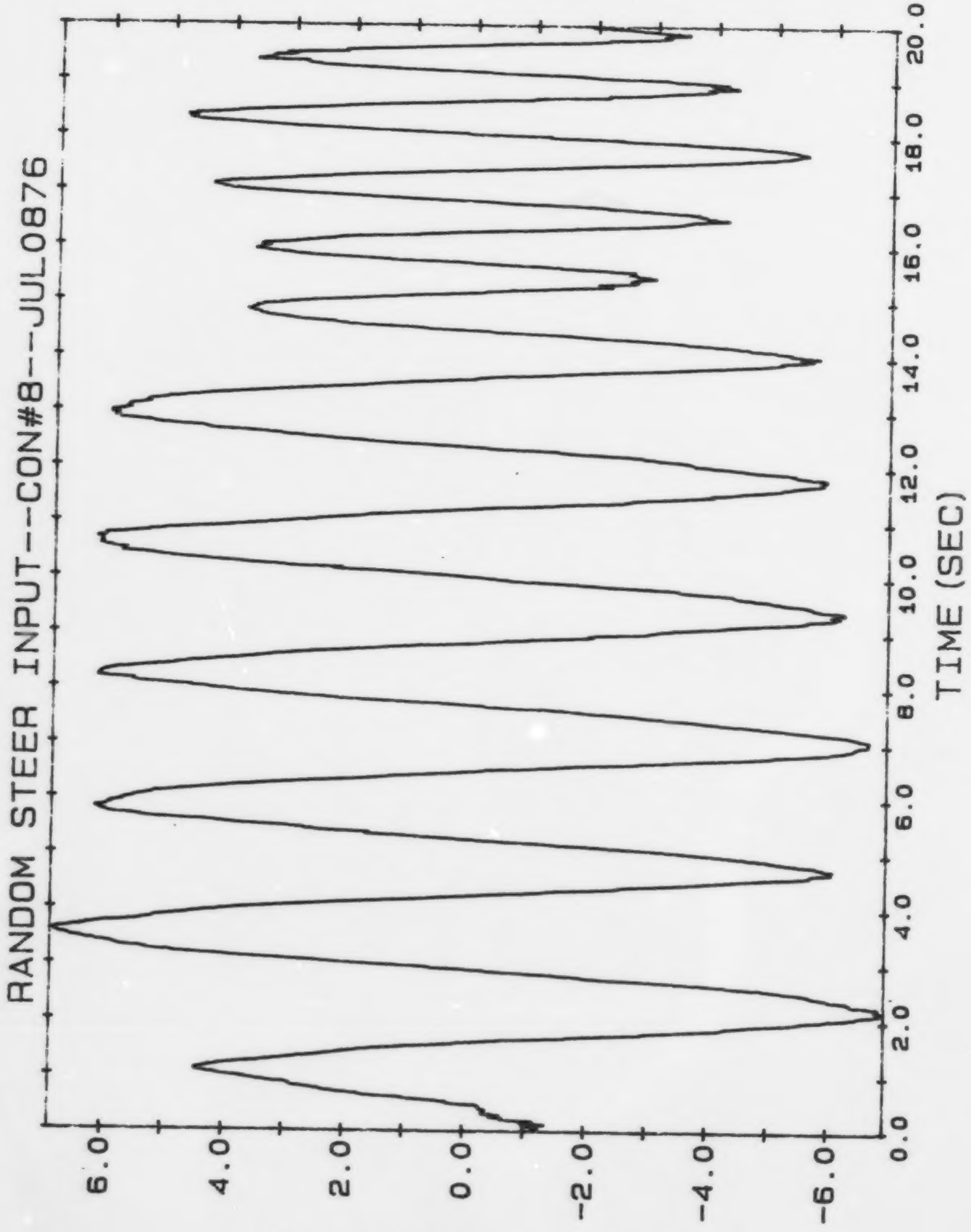

RANDOM STEER INPUT--CON#8--JUL0876
TIME (SEC)
0.0
2.0
4.0
6.0
8.0
10.0
12.0
14.0
16.0
18.0
20.0
YAW RATE (DEG/SEC)
6.0
4.0
2.0
0.0
-2.0
-4.0
-6.0

FIGURE 7.16

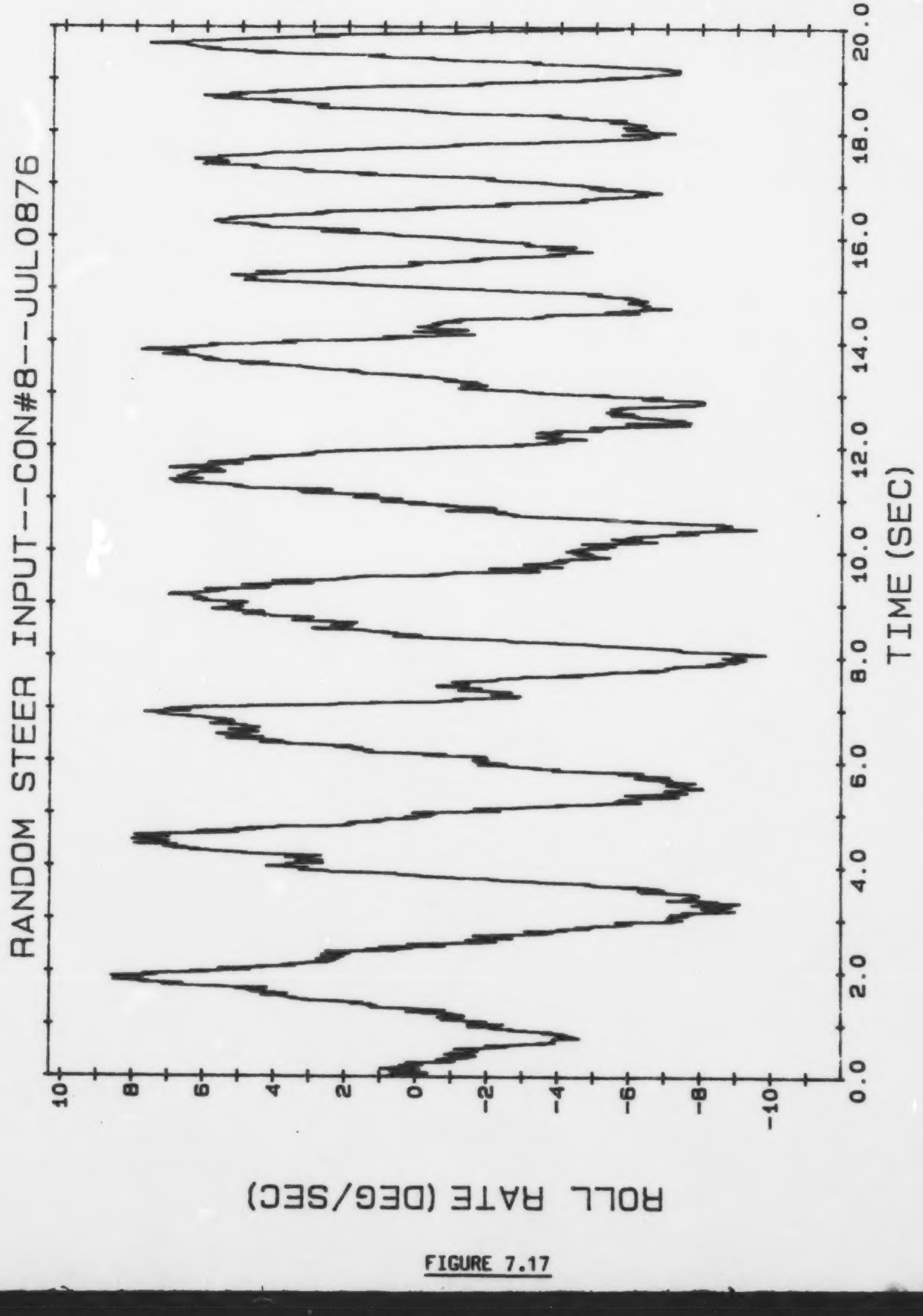

RANDOM STEER INPUT--CON#8--JUL0876
ROLL RATE (DEG/SEC)
TIME (SEC)
10
8
6
4
2
0
-2
-4
-6
-8
-10
0.0
2.0
4.0
6.0
8.0
10.0
12.0
14.0
16.0
18.0
20.0

FIGURE 7.17

APPENDIX 1

Detailed Breakdown of Configurations

Sixteen configurations are described in section 2.0. From these configurations, eight unique parametric changes can be itemized.

1) Depressurizing the tag wheels

Tag wheels were depressurized by operating two three-way air valves located in the right rear engine compartment. When the valve levers were vertical, the tags were pressurized; when the valve levers were horizontal, the tags were depressurized.

2) Chaining-up the tags

Tag wheels were first depressurized before chaining up. The test bus was driven in reverse such that a tag wheel was raised on the "run-up" block (supplied with bus). Using a length of chain and a nut and bolt, the tag was chained in place and supported by a chain hook (supplied on the tag) and the rear suspension bogie structure. The remaining side followed the same procedure. This left both tag wheels with approximately 10 mm clearance between the tire surface and the road.

3) Changing tire pressures

Standard tires were set to:

front	100 psi
drive	80 psi
tag	70 psi

Low profile tires were set to:

all tires at 100 psi

and for one test:

all tires at 70 psi

4) Changing tire type

The two tire types were:

Standard tires - Michelin XZA 12R22.5
Low Profile tires - Michelin Pilot XZA-1 275/80R

5) Disconnect anti-roll bar

Four bushing retainer clamps hold the anti-roll bar to the bus body. To disconnect the anti-roll bar, these four clamps were removed. A piece of rope was then tied between the body and the anti-roll bar to prevent the bar from scrubbing the ground.

6) Disconnect shock absorbers

The two rear drive shock absorbers were selected for removal.

7) Adding weight (uniform weight distribution)

APPENDIX 2

Location of Transducers in Global Coordinate System

The Coordinate axis used is a right handed coordinate system positioned centrally in the bus body at ground level with the x-axis pointing to the front of the bus; the y-axis pointing to the right side; and the z-axis pointing down.

Accelerometers			(Location in Metres)
± 2g Model #SA102	Front Right Seat Post		(4.495,.584,-1.270)
± 2g Model #SA102	c.g. and on floor		(-.500,0,-1.308)
± 2g Model #SA102	Rear Left Seat Post		(-4.350,-.584,-1.346)
± 1g Model	On Tripod (Lateral)		(-.500,0,-2.375)
± 1g Model	On Tripod (Longitudinal)		(-.500,0,-2.375)
± 5g Model #SA102	Front Axle		(4.343,-0.075,-.461)
± 5g Model #SA102	Drive Axle		(-2.830,-0.148,-0.827)
± 100mm Model D1603	Front Right Drive Shock	body	(-2.564,.575,-0.947)
		axle	(-2.564,.575,-0.397)
± 100mm Model D1603	Front Left Drive Shock	body	(-2.564,.575,-0.947)
		axle	(-2.564,.575,-0.397)
± 100mm Model SE373/100	Right Front Axle	body	(4.508,.547,-1.160)
		axle	(4.508,.547,-0.779)

Displacement Transducers		(Location in Metres)	
± 100mm Model SE373/100	Left Front Axle	body	(4.508,.570,-1.160)
		axle	(4.508,.570,-0.779)
± 100mm Model	Tie rod to axle	tie rod	(4.163,0.0,-.564)
		axle	(4.211,-.394,-.564)
± 50mm Model #D1302	Right longitudinal	body	(4.927,.547,-.779)
	(Front axle to body)	axle	(4.610,.547,-.779)
± 50mm Model #D1302	Left longitudinal	body	(4991,-.570,-.779)
	(Front axle to body)	axle	(4597,-.570,-.779)

Other Transducers

Rate Gyro	At c.g. on bus floor	(-.500,0,1.333)
Slip/Roll trolley	Behind Drive Axle	(-3.150,0,-.323)

APPENDIX 3

On-Board Data Acquisition System

On-board data acquisition consisted primarily of three unique hardware devices.

1) Signal Conditioning Box
2) 14 channel analog FM tape recorder
3) Zenith Microcomputer with WAVEPAK software and hardware installed.

Description of Items

1) Signal Conditioning Box
(Designed and Built In-house)

The function of the signal conditioning box was to provide amplification and offset to signals coming from transducers.

Maximum offset is ± 5V; range of gains is 0-22; and maximum voltage output is ± 10V. Sixteen channels are available from the signal conditioning box. (On-board the test bus, the signal conditioning box was used only on the trolley slip and roll signals).

2) 14 Channel Analog FM Tape Recorder
(Sangamo Sabre VI Magnetic Tape Recorder)

The FM tape recorder was used as the primary recording device from this test program. Signals were accepted either from the signal conditioning box or directly from the transducers.

The tape recorder has 14 data channels with one voice side channel. Information was recorded on a 1 inch wide magnetic tape at a speed of $3\frac{3}{4}$ ips.

The tape recorder Record boards were set-up from the following inputs.

Tape Channel	Maximum Voltage Range (V)
1	± 6
2	± 6
3	± 6
4	± 6
5	± 6
6	± 4
7	± 4
8	± 4
9	± 4
10	± 4
11	± 4
12	± 4
13	± 4
14	± 6

3) Zenith Microcomputer

A Zenith Microcomputer was leased from Queen's University, Mechanical Engineering. Installed in the microcomputer was an FFT (Fast Fourier Transform) signal analyzer package known as WAVEPAK (CSI-Computational Systems Incorporated). The package was used on-board the test bus for a "glance" at data. This provided some indication "in the field" as to whether transducers were operating correctly and/or the correct test data was being achieved.

WAVEPAK functioned as a dual channel FFT signal analyzer. WAVEPAK hardware consisted of Analog signal conditioning hardware (for filtering and amplification) and 12 bit Digital data acquisition hardware.

Full scale input range was adjustable from ± 0.001V to ± 10.0V.

Maximum sampling rate was 220,000 samples per second for a single channel (110,000 when dual channels were used).

ATTACHMENT A-2

SIMULATION PROGRAM
DOCUMENTATION

Report No.: 449-625-001

Issue No.: 1

Copy No.:

B. Whitten
Originator

Project Engineer

Date:

Project Manager

Date: 87/08/21

A-2 SIMULATION PROGRAM DOCUMENTATION

1.0 INTRODUCTION

The computer simulation package to examine bus suspension system performance is structured as a group of modules or programs which are run separately in a sequential manner. Data is transferred between modules using computer files, created and saved in disc storage. All modules are written in Fortran-77 programming language, and the following documentation describes the implementation on a Hewlett Packard 1000, Model A900 mini-computer.

The structure of the simulation modules and supporting data files is shown in Figure 1. A brief description of the principal modules is given in Table 1. The use of the simulation package is best described by example, and the procedure followed to model and examine the Test Bus suspension system will be presented here.

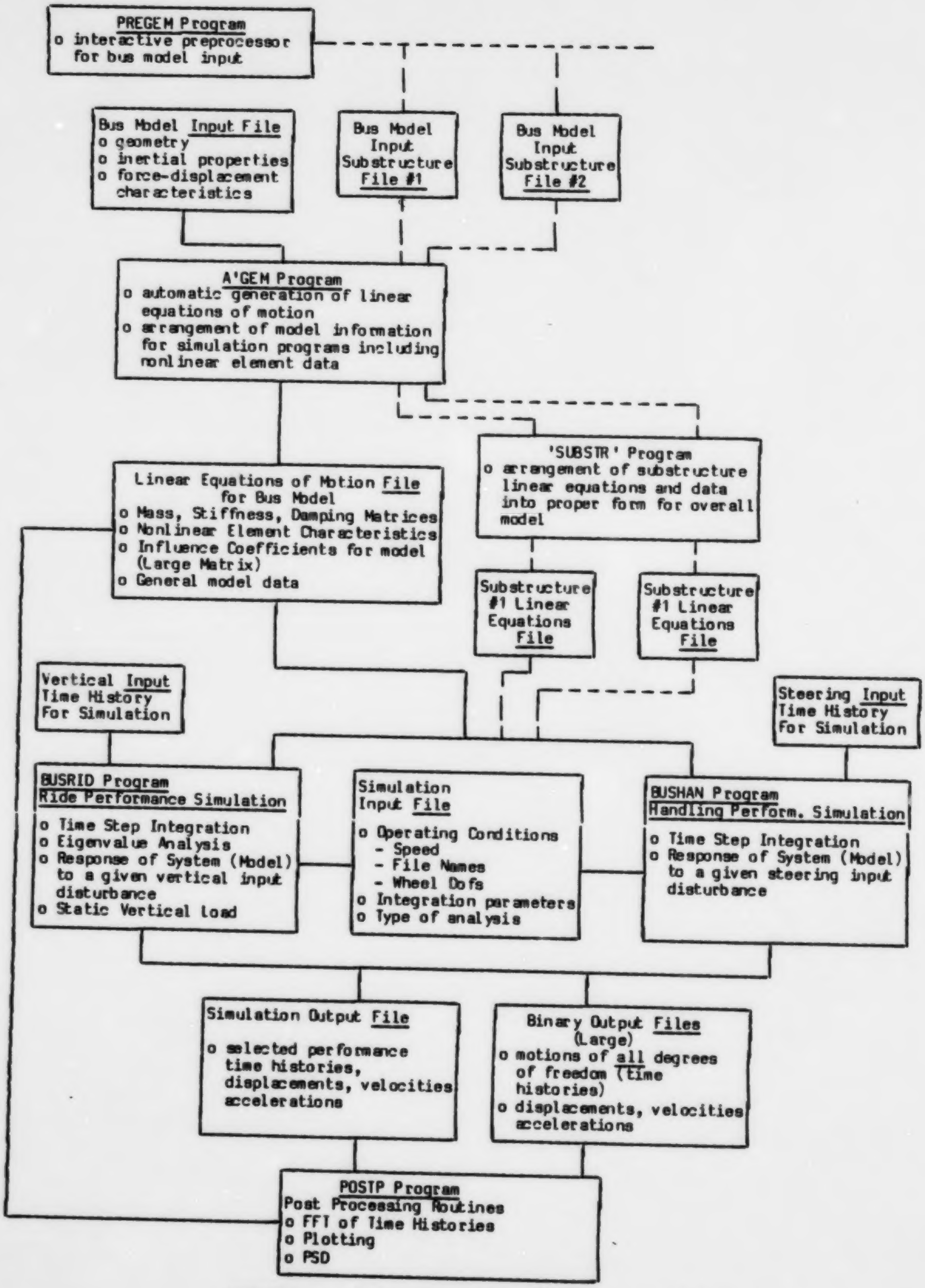

PREGEM Program
o interactive preprocessor for bus model input
Bus Model Input File
o geometry
o inertial properties
o force-displacement characteristics
Bus Model Input Substructure File #1
Bus Model Input Substructure File #2
A'GEM Program
o automatic generation of linear equations of motion
o arrangement of model information for simulation programs including nonlinear element data
Linear Equations of Motion File for Bus Model
o Mass, Stiffness, Damping Matrices
o Nonlinear Element Characteristics
o Influence Coefficients for model (Large Matrix)
o General model data
'SUBSTR' Program
o arrangement of substructure linear equations and data into proper form for overall model
Substructure #1 Linear Equations File
Substructure #1 Linear Equations File
Vertical Input Time History For Simulation
Steering Input Time History For Simulation
BUSRID Program
Ride Performance Simulation
o Time Step Integration
o Eigenvalue Analysis
o Response of System (Model) to a given vertical input disturbance
o Static Vertical Load
Simulation Input File
o Operating Conditions
- Speed
- File Names
- Wheel Dofs
o Integration parameters
o Type of analysis
BUSHAN Program
Handling Perform. Simulation
o Time Step Integration
o Response of System (Model) to a given steering input disturbance
Simulation Output File
o selected performance time histories, displacements, velocities accelerations
Binary Output Files (Large)
o motions of all degrees of freedom (time histories)
o displacements, velocities accelerations
POSTP Program
Post Processing Routines
o FFT of Time Histories
o Plotting
o PSD

FIGURE 1. Block Diagram of Simulation Modules

TABLE 1: Principal Simulation Modules

Module Name	Description	Input Data Files	Output Data Files
PREGEM	An interactive preprocessing program which simplifies the creation of an input data file (the suspension model) for the A'GEM module. Menus are used to prompt the user for data. The output is a properly formed input file for A'GEM containing all data describing the suspension model.	None	Bus suspension system description (model) file, for input to A'GEM
A'GEM	A program which derives the linear equations of motion (mass, stiffness and damping matrices) governing a three-dimensional multibody suspension system by working with a description of the system (model) contained in a simple input file. The program also arranges data describing nonlinear suspension elements into a form suitable for use in modules BUSRID and BUSHAN.	Bus suspension system model description file, giving inertial properties, geometry and force-velocity-displacement characteristics of all components of a rigid body suspension model.	Large binary file for the suspension system, containing the mass, damping and stiffness matrices, the influence coefficients (effect of each degree-of-freedom on the deflection of each suspension element), nonlinear suspension component data, and all geometrical and inertial properties required for the suspension performance modules.
SUBSTR	A program (used only for suspension system models comprised of one or more substructures) which produces equivalent stiffness and damping matrices for each substructure of the system.	A file, output from A'GEM, containing all matrices and information regarding a given substructure of the suspension system.	A binary file containing equivalent stiffness and damping matrices for a substructure.

TABLE 1: Principal Simulation Modules (Cont'd)

Module Name	Description	Input Data Files	Output Data Files
BUSRID	A ride performance program which computes the response of the suspension system model to given vertical inputs at the wheels. The equations of motion are arranged and integrated in a time-step fashion to give the displacement, velocity and acceleration of each degree-of-freedom in the system. Additional options include the response of the system to a static vertical force of 1 g acceleration (gravity), and an eigenvalue analysis of the linear system for characteristic frequencies and modes.	1. Program control file, specifying response option required, wheels to be given vertical input, integration parameters, type of vertical input, vehicle speed, etc. 2. Large suspension system file output from A'GEM. 3. If specified in the control file, a tabulated file of vertical inputs as a function of time.	1. A general output file, confirming inputs and giving selected results. 2. A large binary file giving the displacement of each DOF at each time step. 3. A large binary file giving the velocity of each DOF at each time step. 4. A large binary file giving the acceleration of each DOF at each time step.
BUSHAN	A handling performance program which determines the response of the suspension system model to given steering (lateral) inputs at the wheels. The equations of motion are arranged and integrated in a time-step manner to give the displacement, velocity and acceleration of each DOF in the system. The angles of each road wheel relative to its corresponding axle are used as principal inputs to the simulation.	1. Program control file, specifying wheels to be steered, static normal forces through each wheel, vehicle speed, type of steering input, integration parameters, etc. 2. Large suspension system file output from A'GEM. 3. If specified, a tabulated file of steering inputs as a function of time.	1. A general output file, confirming inputs and giving selected results. 2. A large binary file giving the displacement of each DOF at each time step. 3. A large binary file giving the velocity of each DOF at each time step. 4. A large binary file giving the acceleration of each DOF at each time step.
POSTP	An interactive post-processing routine which computes motions, forces, etc. at any given location within the suspension system as chosen by the user. The selection may include the displacement, velocity and force through any suspension element. Particular requirements for a given suspension system may be incorporated into the computer code.	1. All output files from either BUSRID or BUSHAN. 2. File output from A'GEM containing all suspension system information.	1. A file containing time histories of selected motions, forces, etc. for plotting purposes.

2.0 MODEL CREATION - INPUT FILE FOR A'GEM

The physical suspension system must first be modelled by the designer in conceptual form as a system of rigid bodies connected by suspension elements. Data must be gathered on the inertial properties of each rigid body, the geometrical location of each element and their interconnections, and the force-displacement-velocity characteristics of each interconnecting element. This information must be incorporated into an input file for A'GEM, and a preprocessor (PREGEM) can be used to assist in this function. The components of the conceptual model used by A'GEM are:

1. Rigid Bodies: these are bodies which have inertial and dimensional properties, and whose motions are the primary dynamic variables. The "Ground" is a special rigid body which is not allowed to move.

2. Reference Frame: the equations-of-motion are derived by allowing each rigid body to move in 3-dimensional space. The description of these motions requires a set of co-ordinate axes for reference, and a body may then have linear plus rotational motion about these axes.

3. The Origin: a point in space which is assigned co-ordinates (0, 0, 0). All positions are specified with respect to this point. The origin is not attached to any rigid body although it is often convenient to choose a location which coincides with an important point on a body such as a centre of mass.

4. Degrees-of-Freedom: the actual motions allowed for the rigid bodies. The user may allow each body to have freedom to move in any number of directions varying from a low of zero to a high of six. A body with no motion is, by definition, the ground. Other bodies may have no user specified degrees-of-freedom if their motion is completely governed by constraints. The degrees-of-freedom are numbered sequentially from 1 to the total number of degrees-of-freedom for the entire system.

5. Springs: these are the elastic elements which connect neighbouring rigid bodies. Each spring is characterized by its stiffness, the positions of its ends, and the bodies to which the ends are attached.

6. Dampers: dampers are analogous to springs with the only differences being that forces within them are proportional to the relative velocity of their end points. In A'GEM, it is assumed that every spring acts in parallel with a damper so that both stiffness and damping values are required for any element which is defined. This reduces the amount of positional data which must be provided. If an element consists of only a spring and does not produce damping forces, the input data simply specifies a damping coefficient of zero.

7. Non-linear Connection Elements: forces within these elements are non-linear functions of the displacement and velocity across the elements. Each element is characterized by the positions of its ends, the bodies to which the ends are attached and short tables of force versus displacement and velocity.

The information is incorporated into an input description file for A'GEM. All numerical data is read in list-directed form so that spacing is not important. Data is in the following order with variable names as they appear in the code:

1. 80 alphanumeric characters
   DUM

   DUM = a comment

   Notes for record 1:

   a. Record 1 must be repeated 3 times.

   b. This record, and subsequent records using the variable DUM, are included for ease in reading this input file. The comments placed here are not used and do not appear in any output files.

2. 80 alphanumeric characters
   HEAD

   HEAD = a descriptive heading

   Note: HEAD will appear in output files.

3. 80 alphanumeric characters
   DUM

   DUM = a comment

   Note: Record 3 must be repeated 3 times.

4. 1 integer
   NB

   NB = the number of rigid bodies

   Note: The following sequence of input data (records 5 and 6 must be repeated NB times).

5. 3 real numbers
   (POSB(J,I),J=1,3)

   POSB(1,I) = the x position of the centre of mass of body I
   POSB(2,I) = the y position of the centre of mass of body I
   POSB(3,I) = the z position of the centre of mass of body I

6. 7 real numbers
(MASS(J,I),J=1,7)

MASS(1,I) = the mass of body I
MASS(2,I) = the amount of inertia about the x-axis of body I
MASS(3,I) = the x-y product of inertia of body I
MASS(4,I) = the x-z product of inertia of body I
MASS(5,I) = the moment of inertia about the y-axis of body I
MASS(6,I) = the y-z product of inertia of body I
MASS(7,I) = the moment of inertia about the z-axis of body I

Note for records 5 and 6:

All products of inertia should be entered with the sign arising from their determination. The negative signs appearing on the off-diagonal terms of the inertia tensor are inserted automatically by A'GEM.

7. 6 integers
(INP(J),J=1,6)

INP(1) = number assigned to the x-degree of freedom of body I
INP(2) = number assigned to the y-degree of freedom of body I
INP(3) = number assigned to the z-degree of freedom of body I
INP(4) = number assigned to the rotational degree-of-freedom about the x-axis of body I
INP(5) = number assigned to the rotational degree-of-freedom about the y-axis of body I
INP(6) = number assigned to the rotational degree-of-freedom about the z-axis of body I

Notes for Record 7:

a. Record 7 must be repeated NB times.

b. The values of INP(J) have the following significance:

INP(J) 0: The value is the number assigned to the degree-of-freedom.
INP(J)=0: The degree-of-freedom does not exist or will be handled as indicated by constraints arising from pinned joints.
INP(J) 0: The degree-of-freedom is actually in the negative sense of the reference axis.
INP(J) 500: The degree-of-freedom will be treated as being governed by pinned constraints but will be constrained to be zero before the final equations of motion are produced.

8. 80 alphanumeric characters
DUM

DUM = a comment

Note: Record 8 must be repeated 3 times.

9. 1 integer
NPE

NPE = the number of linear point connection elements.

Notes:

a. The following sequence of input data (records 10, 11, 12 and 13) must be repeated NPE times.

b. NPE counts the number of unique point elements. That is, if there are many locations which use identical elements there will only be one point element. The locations are specified after specifying the stiffness and damping values. This avoids problems associated with repeating the same input many times.

10. 6 real numbers
(TK(J),J=1,6)

TK(1) = stiffness in the x-direction
TK(2) = stiffness in the y-direction
TK(3) = stiffness in the z-direction
TK(4) = torsional stiffness about the x-axis
TK(5) = torsional stiffness about the y-axis
TK(6) = torsional stiffness about the z-axis

11. 6 real numbers
(TC(J),J=1,6)

TC(1) = damping coefficient in the x-direction
TC(2) = damping coefficient in the y-direction
TC(3) = damping coefficient in the z-direction
TC(4) = torsional damping coefficient about the x-axis
TC(5) = torsional damping coefficient about the y-axis
TC(6) = torsional damping coefficient about the z-axis

12. 1 integer
NTHS

NTHS = the number of point elements of this type

Note: Record 13 must be repeated NTHS times.

13. 2 integers, 3 real numbers
IN1,IN2,P1,P2,P3

IN1 = body number to which one end of the element is connected
1N2 = body number to which the other end of the element is connected
P1 = x-position of the element
P2 = y-position of the element
P3 = z-position of the element

Note: The ground is body number 0.

14. 80 alphanumeric characters
DUM

DUM = a comment

Note: Record 14 must be repeated 3 times.

15. 1 integer
NS2

NS2 = the number of linear directed connection elements

Note: Records 16, 17 and 18 must be repeated NS2 times.

16. 1 integer, 3 real numbers
NEND(1,I),(POSS(J,I),J=1,3)

NEND(1,I) = body number to which one end of the element is connected
POSS(1,I) = x-position of this connection
POSS(2,I) = y-position of this connection
POSS(3,I) = z-position of this connection

17. 1 integer, 3 real numbers
NEND(2,I),(POSS(J,I),J=4,6)

NEND(2,I) = body number to which the other end of the element is connected
POSS(4,I) = x-position of this connection
POSS(5,I) = y-position of this connection
POSS(6,I) = z-position of this connection

18. 2 real numbers
KINP(I),CINP(I)

KINP(I) = stiffness of the element
CINP(I) = damping coefficient of the element
Note: Directed elements must have non-zero lengths

19. 80 alphanumeric characters
DUM

DUM = a comment

Note: Record 19 must be repeated 3 times.

20. 1 integer
NSNON

NSNON = the number of nonlinear directed connection elements

Note: Records 21 to 27 must be repeated NSNON times.

21. 1 integer, 3 real numbers
NEND(1,I),(POSS(J,I),J=1,3)

NEND(1,I) = body number to which one end of the element is connected
POSS(1,I) = x-position of this connection
POSS(2,I) = y-position of this connection
POSS(3,I) = z-position of this connection

22. 1 integer, 3 real numbers
NEND(2,I),(POSS(J,I),J=4,6)

NEND(2,I) = body number to which the other end of the element is connected
POSS(4,I) = x-position of this connection
POSS(5,I) = y-position of this connection
POSS(6,I) = z-position of this connection

23. 2 integers
NPX(I),NPV(I)

NPX(I) = the number of data points used to define the force through the element as a function of displacement
NPV(I) = the number of data points used to define the force through the element as a function of velocity

24. NPX(I) real numbers; omit if NPX(I)=0
(XN(I,J),J=1,NPX(I))

XN(I,J) = values of displacement, in monotonically increasing order, at which the force (as a function of displacement) through the element is defined

25. NPX(I) real numbers; omit if NPX(I)=0
(FXN(I,J),J=1,NPX(I))

FXN(I,J) = values of force corresponding to the displacements XN(I,J)

26. NPV(I) real numbers; omit if NPV(I)=0
(VN(I,J),J=1,NPV(I))

VN(I,J) = values of velocity, in monotonically increasing order, at which the force (as a function of velocity) through the element is defined

27. NPV(I) real numbers; omit if NPV(I)=0
(FVN(I,J),J=1,NPV(I))

FVN(I,J) = values of force corresponding to the velocities VN(I,J)

28. 80 alphanumeric characters
DUM

DUM = a comment

Note: Record 28 must be repeated 3 times.

29. 1 integer
NPIN

NPIN = the number of pinned joints in the system

Note: Record 30 must be repeated NPIN times.

30. 5 integers, 3 real numbers
NB1,NB2,IDX,IDY,IDZ,PX,PY,PZ

NB1 = body number of one of the pinned bodies
NB2 = number of the other pinned body
IDX = 1 for a pin in the x-direction 0 otherwise
IDY = 1 for a pin in the y-direction 0 otherwise
IDZ = 1 for a pin in the z-direction 0 otherwise
PX = x position of the pin
PY = y position of the pin
PZ = z position of the pin

Note: pins must be parallel to one of the reference axes. Only one of IDX, IDY and IDZ may be equal to one, the others must be zero.

2.1 TEST BUS SUSPENSION SYSTEM MODEL

The computer model for the 3-Axle Test Bus in given as an example to demonstrate a representative input file required for A'GEM. All dimensions and parameters are expressed in SI units, and the right-handed reference frame is chosen as shown in Figure 2 with the "x" longitudinal direction pointing forward, the "y" lateral direction pointing to the right, and the "z" vertical direction pointing downwards. The origin is located at ground level at the intersection of the bus body longitudinal and lateral centerline.

The suspension system model is also shown in Figure 2. The bus body is supported vertically at each axle by air springs, with a damper (shock absorber) in parallel with each spring. The front beam axle is positioned longitudinally with respect to the body by upper and lower trailing links, and laterally by a panhard rod. The beam drive axle is located by two upper links, diagonal so as to provide both lateral and longitudinal support. An anti-roll torsion bar is also connected between the bus body and the drive axle to provide additional roll stiffness. An independent tag axle (one each side), pivoting from the bus body about a support pin complete the suspension arrangement. A total of 8 tires (2 front, 4 drive and 2 tag tires) support the vehicle on the roadway.

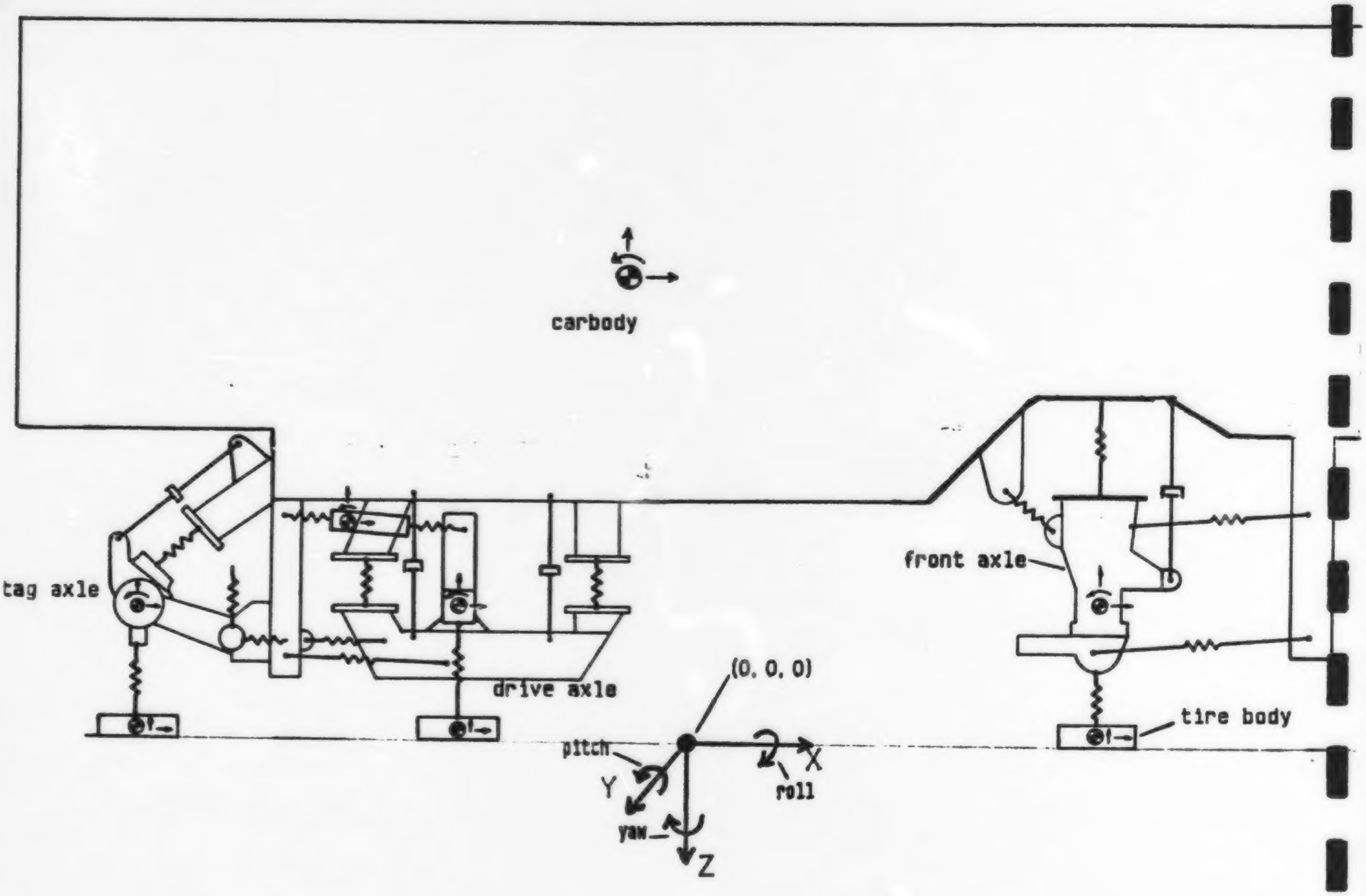

carbody
tag axle
drive axle
front axle
tire body
(0, 0, 0)
pitch
Y
X
roll
yaw
Z

FIGURE 2 - BUS MODEL AND REFERENCE CO-ORDINATE SYSTEM

The computer model is comprised of:

- 13 rigid bodies
- 65 linear spring-damper combinations, pointing in either the principal x, y or z directions, or torsionally about these directions.
- 9 linear spring-damper combinations, pointing in a diagonal direction.
- 8 non-linear dampers (shock absorbers), pointing in a diagonal direction.

The modelling procedure, in a step-by-step fashion, is as follows:

1. Identify and assign numbers to the rigid bodies in the system. The numbering system is arbitrary, but necessary. The bodies will retain this numbering in the input data file for A'GEM.

2. Choose a right-handed reference frame and a suitable origin (Figure 2). This establishes six reference directions.

3. Specify degrees-of-freedom for each rigid body in the system by assigning unique numbers. Motions to be ignored are given the value zero. Each rigid body may be allowed up to 6 degrees-of-freedom in the reference directions, 3 linear (longitudinal, lateral, vertical) and 3 angular (roll, pitch, yaw). Since the bus suspension model is to be subjected to both vertical (ride) and lateral (handling) inputs, most bodies are given all 6 degrees-of-freedom.

4. Locate the centres of mass of the rigid bodies with respect to the origin. To do this, positive directions are as specified by the reference axes.

5. Determine the mass of each body in units appropriate for the analysis. A'GEM has no restriction on units. The user is responsible for using a consistent set. The moments and products of inertia about the reference directions are also required if angular degrees-of-freedom are indicated.

6. Locate the springs and dampers and determine to which bodies they are connected. For ease of input, these force producing elements are categorized as either "linear point connection elements", "linear directed connection elements" or "nonlinear directed connection elements".

   A linear point connection element connects two bodies at a single point in space and has stiffness and damping coefficients in all six reference directions; three linear and three torsional. These elements are useful for modelling blocks of rubber and bushings, and are very convenient in describing most connection elements. In the Test Bus Model, 9 point connection elements are used to model 65 spring/damper combinations in the various reference directions.

Directed connection elements, both linear and nonlinear, connect two bodies at separate points in space and can produce forces only along their axes. Examples of linear directed elements in the Bus Model are the trailing links and the tag axle air springs. Examples of nonlinear directed connection elements are the shock absorbers.

A listing of the input file to A'GEM describing the Test Bus Model is given in Table 2. Sequential line numbers have been added to the left of the listing only to assist in this documentation. The format of the file is as described previously, with 5 principal sections:

Section A: Rigid Body Data (Line nos. 5 to 47)
B: Point Connection Element Data (line nos. 48 to 99)
C: Linear Directed Element Data (line nos. 100 to 130)
D: Nonlinear Directed Element Data (line nos. 131 to 174)
E: Pinned Joint Data (line nos. 175 to 178)

Comments, which follow an asterisk on many lines, can and have been added to the input file to assist in the management of the model description.

When this file is input to A'GEM, the format of the data is initially checked, rearranged and reprinted in a "check" file. A listing of this check file appears in Table 3, and is generally used as a guide during the remainder of the simulation process because of its' convenient format. Rigid body numbers and their degrees-of-freedom are confirmed, and "join numbers" are assigned to every connection in the system for future reference.

TABLE 2 TEST BUS MODEL DESCRIPTION FILE. INPUT TO A'GEM

File: /SUSPENSION/BU1ENL

```
Line No
   1      *
   2      *       A'GEM Input File
   3      *
   4      BUS MODEL #1E - Test Bus, Loaded, Standard Tires, Nonlinear Shocks
   5      *
   6      *  SECTION A - Rigid Body Data
   7      *
   8        13
   9      -0.500000   .0203000  -1.143000    * Bus Body
  10       13750.00   24932.000  .0000E+00  .0000E+00  193724.00  .0000E+00  183879.00
  11      4.3430000    0.0000  -.5020000    * Front Axle
  12       916.0       923.0000  .0000E+00  .0000E+00  8.2000000  .0000E+00   921.0000
  13      4.3430000  1.0920000   -.502000    * Front Contact Patch, Right
  14       10.00       .0000E+00  .0000E+00  .0000E+00  .0000E+00  .0000E+00  .0000E+00
  15      4.3430000  -1.092000   -.502000    * Front Contact Patch, Left
  16       10.00       .0000E+00  .0000E+00  .0000E+00  .0000E+00  .0000E+00  .0000E+00
  17      -2.896000    0.0000  -.5020000    * Drive Axle
  18       1806.       930.00000  .0000E+00  .0000E+00  52.300000  .0000E+00  945.00000
  19      -2.896000   .9940000   -.502000    * Drive Contact Patch, Right
  20       10.00       .0000E+00  .0000E+00  .0000E+00  .0000E+00  .0000E+00  .0000E+00
  21      -2.896000  -.9940000   -.502000    * Drive Contact Patch, Left
  22       10.00       .0000E+00  .0000E+00  .0000E+00  .0000E+00  .0000E+00  .0000E+00
  23      -3.237000   .3350000  -.8420000     * A-Link, Right
  24       25.40       1.1500000  -1.395600  .0000E+00  1.7030000  .0000E+00  2.9700000
  25      -3.237000  -.3350000  -.8420000     * A-Link, Left
  26       25.40       1.1500000  +1.395600  .0000E+00  1.7030000  .0000E+00  2.9700000
  27      -4.102000  +0.9330000  -.5020000     * Tag Axle, Right
  28       275.0       9.5000       .0000E+00  .0000E+00  4.0000000  .0000E+00  11.500
  29      -4.102000  -0.933000  -.5020000     * Tag Axle, Left
  30       275.0       9.5000       .0000E+00  .0000E+00  4.0000000  .0000E+00  11.500
  31      -4.125000  1.0710000  -.5020000     * Tag Contact Patch, Right
  32       10.00       .0000E+00  .0000E+00  .0000E+00  .0000E+00  .0000E+00  .0000E+00
  33      -4.125000  -1.071000  -.5020000     * Tag Contact Patch, Left
  34       10.00       .0000E+00  .0000E+00  .0000E+00  .0000E+00  .0000E+00  .0000E+00
  35          1     2     3     4     5     6
  36          7     8     9    10    11    12
  37         31    32    33     0     0     0
  38         34    35    36     0     0     0
  39         13    14    15    16    17    18
  40         37    38    39     0     0     0
  41         40    41    42     0     0     0
  42         49    50    51    52    53    54
  43         55    56    57    58    59    60
  44         19    20    21    22    23    24
  45         25    26    27    28    29    30
  46         43    44    45     0     0     0
  47         46    47    48     0     0     0
  48      *
  49      *  SECTION B - Point Element Data
  50      *
  51         9
  52       8856545.0     876608.      9.8750E+05  0.00000     0.00000      0.0000
  53       2.00E4        2.00E4       2.000E2     0.00000     0.00000      0.0000
  54         2        * Front Tires
  55         2     3     4.343       1.092        -.502
  56         2     4     4.343      -1.092        -.502
  57       8856545.0     876608.      8.7440E5    0.00000     0.00000      0.0000
  58       2.00E4        2.00E4       2.000E2     0.00000     0.00000      0.0000
  59         4        * Drive Tires
  60         5     6      -2.896     1.168        -.502
  61         5     6      -2.896     .819         -.502
```

TABLE 2: TEST BUS MODEL DESCRIPTION FILE. INPUT TO A'GEM (CONT'D)

```
 62        5    7      -2.896     -1.168       -.502
 63        5    7      -2.896     -.819        -.502
 64    8856545.0     876608.      8.0310E5     0.00000     0.00000      0.0000
 65    2.00E4        2.00E4       2.000E2      0.00000     0.00000      0.0000
 66        2       * Tag Tires
 67        10   12     -4.102      1.071       -.502
 68        11   13     -4.102     -1.071       -.502
 69     0.00000      0.00000      1.8000E+05   0.00000     0.00000      0.0000
 70    0.00000       0.00000      0.00000      0.00000     0.00000      0.0000
 71        2        * Front Air Springs
 72        1    2       4.343      0.591       -.700
 73        1    2       4.343     -0.591       -.700
 74     0.00000      0.00000      2.0000E+05   0.00000     0.00000      0.0000
 75    0.00000       0.00000      0.00000      0.00000     0.00000      0.0000
 76        2        * Drive Air Springs, Right Side
 77        1    5      -2.375      0.464       -.508
 78        1    5      -3.252      0.464       -.508
 79     0.00000      0.00000      2.0000E+05   0.00000     0.00000      0.0000
 80    0.00000       0.00000      0.00000      0.00000     0.00000      0.0000
 81        2        * Drive Air Springs, Left Side
 82        1    5      -2.375     -0.464       -.508
 83        1    5      -3.252     -0.464       -.508
 84    .26270E+08   .43784E+07   .26270E+08   23540.      2134.0       23540.
 85    1.00E+03      5.00E+02     1.00E+03     10.          10.          10.
 86        4       * Bushings, A-Links
 87        8    5 -2.932000    .0810000   -.8270000
 88        8    1 -3.541000    .5870000   -.8560000
 89        9    5 -2.932000   -.0810000   -.8270000
 90        9    1 -3.541000   -.5870000   -.8560000
 91    0.000000     0.000000     0.000000     27487.000   0.000000     0.000000
 92    0.00         0.0000       0.000000     0.0000000   0.000000     0.000000
 93        1       * Roll Bar
 94        1    5      -3.3160     0.00000     -.293000
 95    .10000E+09   .10000E+09   .10000E+09   .10000E+00  .00000E+00   .10000E+00
 96    .00000E+00   .00000E+00   .00000E+00   .00000E+00  .00000E+00   .00000E+00
 97        2       * Tag Axle Pivots
 98        1   10 -3.741000    .7150000   -.3830000
 99        1   11 -3.741000   -.7150000   -.3830000
100   *
101   *  SECTION C - Linear Directed Element Data
102   *
103        9
104        1 4.1240000  -.4640000  -.6730000
105        2 4.2000000   .6540000  -.6060000
106     6.5675E+06   1.00E+04    * Front Panhard Rod
107        1 4.9460000   .3700000  -.6600000
108        2 4.3430000   .3950000  -.6060000
109     1.3135E+07  1.00E+04    * Front, Upper Trailing Link, Right
110        1 4.9460000   .6060000  -.3050000
111        2 4.3430000   .6320000  -.2510000
112     1.3135E+07  1.00E+04   * Front, Lower Trailing Link, Right
113        1 4.9460000  -.3700000  -.6600000
114        2 4.3430000  -.3950000  -.6060000
115     1.3135E+07  1.00E+04    * Front, Upper Trailing Link, Left
116        1 4.9460000  -.6060000  -.3050000
117        2 4.3430000  -.6320000  -.2510000
118     1.3135E+07  1.00E+04    * Front, Lower Trailing Link, Left
119        1 -3.541000   .5870000  -.3230000
120        5 -2.932000   .5870000  -.3000000
121     1.3135E+07  1.00E+04    * Drive Axle Leading Link, Right
122        1 -3.541000  -.5870000  -.3230000
123        5 -2.932000  -.5870000  -.3000000
124     1.3135E+07  1.00E+04    * Drive Axle Leading Link, Left
125        1 -3.886000   .6990000  -.8280000
```

TABLE 2: TEST BUS MODEL DESCRIPTION FILE, INPUT TO A'GEM (CONT'D)

```
    10 -4.000000    .6990000  -.6120000
  90000.    00000.        * Tag Axle Air Spring, Right
     1 -3.886000  -.6990000  -.8280000
    11 -4.000000  -.6990000  -.6120000
  90000.    00000.        * Tag Axle Air Spring, Left
 *
 *  SECTION D - Nonlinear Directed Element Data
 *
     8
      1 4.55500    .67600     -0.990000
      2 4.55500    .62900     -.430000
     0     8                  * Front Shock Absorber, Right
  -.080 -.008 -.005 .005 .008  .024 .040 .080
   -555. -175. -100.  100.  150.  300. 515. 1540.
      1 4.10000    -.67300    -0.99000
      2 4.10000    -.59700    -.43000
     0     8                  * Front Shock Absorber, Left
  -.080 -.008 -.005 .005 .008  .024 .040 .080
   -555. -175. -100.  100.  150.  300. 515. 1540.
      1  -2.564        0.575      -0.968
      5  -2.564        0.575      -.365
     0     9                  * Drive Shock Absorber, Right Leading
  -.080  -.040  -.008 -.005  .005  .008 .024 .040  .080
   -220.  -162.  -138. -100.  100.  150. 300. 515. 1540.
      1  -2.564       -0.575      -0.968
      5  -2.564       -0.575      -.365
     0     9                  * Drive Shock Absorber, Left Leading
  -.080  -.040  -.008 -.005  .005  .008 .024 .040  .080
   -220.  -162.  -138. -100.  100.  150. 300. 515. 1540.
      1  -3.069        0.575      -0.968
      5  -3.069        0.575      -.365
     0     9                  * Drive Shock Absorber, Right Trailing
  -.080  -.040  -.008 -.005  .005  .008 .024 .040  .080
   -220.  -162.  -138. -100.  100.  150. 300. 515. 1540.
      1  -3.069       -0.575      -0.968
      5  -3.069       -0.575      -.365
     0     9                  * Drive Shock Absorber, Left Trailing
  -.080  -.040  -.008 -.005  .005  .008 .024 .040  .080
   -220.  -162.  -138. -100.  100.  150. 300. 515. 1540.
      1 -3.74400     .78100     -1.00000
     10 -4.20000     .78100     -.72200
     0     12                 * Tag Axle Shock Absorber, Right
  -.080 -.064 -.040 -.024 -.008 -.005  .005  .008 .020 .040  .060  .080
   -220. -200. -162. -156. -138. -100.  100.  138. 175. 250.  370.  500.
      1 -3.74400  -.78100     -1.00000
     11 -4.20000  -.78100     -.72200
     0     12                 * Tag Axle Shock Absorber, Left
  -.080 -.064 -.040 -.024 -.008 -.005  .005  .008 .020 .040  .060  .080
   -220. -200. -162. -156. -138. -100.  100.  138. 175. 250.  370.  500.
 *
 *  Pinned Joint Data
 *
     0
```

TABLE 3 "CHECK" FILE OF TEST BUS MODEL, PRINTED BY A'GEM

```
A'GEM CHECK FILE - 10:42 AM  THU.,  4  JUN., 1987

       Input File: BUIENL
      Output File: BUIENL01.GEM

BUS MODEL #1E - Test Bus, Loaded, Standard Tires, Nonlinear Shocks

   THERE ARE  13 RIGID BODIES

Body        C.G. POSITION              MASS       IXX       IXY      IXZ      IYY       IYZ      IZZ
 No
  1    -.5000    .0203  -1.1430   13750.0   24932.0      0.0      0.0  193724.0      0.0  183879.0   Bus Body
  2    4.3430   0.0000   -.5020     916.0     923.0      0.0      0.0       8.2      0.0     921.0   Front Axle
  3    4.3430   1.0920   -.5020      10.0       0.0      0.0      0.0       0.0      0.0       0.0   Front Contact Patch, Right
  4    4.3430  -1.0920   -.5020      10.0       0.0      0.0      0.0       0.0      0.0       0.0   Front Contact Patch, Left
  5   -2.8960   0.0000   -.5020    1806.0     930.0      0.0      0.0      52.3      0.0     945.0   Drive Axle
  6   -2.8960    .9940   -.5020      10.0       0.0      0.0      0.0       0.0      0.0       0.0   Drive Contact Patch, Right
  7   -2.8960   -.9940   -.5020      10.0       0.0      0.0      0.0       0.0      0.0       0.0   Drive Contact Patch, Left
  8   -3.2370    .3350   -.8420      25.4       1.2     -1.4      0.0       1.7      0.0       3.0   A-Link, Right
  9   -3.2370   -.3350   -.8420      25.4       1.2      1.4      0.0       1.7      0.0       3.0   A-Link, Left
 10   -4.1020    .9330   -.5020     275.0       9.5      0.0      0.0       4.0      0.0      11.5   Tag Axle, Right
 11   -4.1020   -.9330   -.5020     275.0       9.5      0.0      0.0       4.0      0.0      11.5   Tag Axle, Left
 12   -4.1250   1.0710   -.5020      10.0       0.0      0.0      0.0       0.0      0.0       0.0   Tag Contact Patch, Right
 13   -4.1250  -1.0710   -.5020      10.0       0.0      0.0      0.0       0.0      0.0       0.0   Tag Contact Patch, Left
   THERE ARE  60 DEGREES OF FREEDOM
Body
  1        1   2   3   4   5   6   Bus Body
  2        7   8   9  10  11  12   Front Axle
  3       31  32  33   0   0   0   Front Contact Patch, Right
  4       34  35  36   0   0   0   Front Contact Patch, Left
  5       13  14  15  16  17  18   Drive Axle
  6       37  38  39   0   0   0   Drive Contact Patch, Right
  7       40  41  42   0   0   0   Drive Contact Patch, Left
  8       49  50  51  52  53  54   A-Link, Right
  9       55  56  57  58  59  60   A-Link, Left
 10       19  20  21  22  23  24   Tag Axle, Right
 11       25  26  27  28  29  30   Tag Axle, Left
 12       43  44  45   0   0   0   Tag Contact Patch, Right
 13       46  47  48   0   0   0   Tag Contact Patch, Left

  THERE ARE   9 POINT CONNECTION ELEMENTS

Join   End1 End2       ----POSITION----             DIRECTION        STIFFNESS     DAMPING
 No
  1      2    3     4.3430   1.0920   -.5020    1  0  0  0  0  0    8.8565E+06   2.0000E+04
  2      .    .       .        .        .       0  1  0  0  0  0    8.7661E+05   2.0000E+04   Front Tires
  3      .    .       .        .        .       0  0  1  0  0  0    9.8750E+05   2.0000E+02
  4      2    4     4.3430  -1.0920   -.5020    1  0  0  0  0  0    8.8565E+06   2.0000E+04
  5      .    .       .        .        .       0  1  0  0  0  0    8.7661E+05   2.0000E+04   Front Tires
  6      .    .       .        .        .       0  0  1  0  0  0    9.8750E+05   2.0000E+02
  7      5    6    -2.8960   1.1680   -.5020    1  0  0  0  0  0    8.8565E+06   2.0000E+04
  8      .    .       .        .        .       0  1  0  0  0  0    8.7661E+05   2.0000E+04   Drive Tires
  9      .    .       .        .        .       0  0  1  0  0  0    8.7440E+05   2.0000E+02
 10      5    6    -2.8960    .8190   -.5020    1  0  0  0  0  0    8.8565E+06   2.0000E+04
 11      .    .       .        .        .       0  1  0  0  0  0    8.7661E+05   2.0000E+04   Drive Tires
 12      .    .       .        .        .       0  0  1  0  0  0    8.7440E+05   2.0000E+02
 13      5    7    -2.8960  -1.1680   -.5020    1  0  0  0  0  0    8.8565E+06   2.0000E+04
 14      .    .       .        .        .       0  1  0  0  0  0    8.7661E+05   2.0000E+04   Drive Tires
 15      .    .       .        .        .       0  0  1  0  0  0    8.7440E+05   2.0000E+02
 16      5    7    -2.8960   -.8190   -.5020    1  0  0  0  0  0    8.8565E+06   2.0000E+04
 17      .    .       .        .        .       0  1  0  0  0  0    8.7661E+05   2.0000E+04   Drive Tires
 18      .    .       .        .        .       0  0  1  0  0  0    8.7440E+05   2.0000E+02
 19     10   12    -4.1020   1.0710   -.5020    1  0  0  0  0  0    8.8565E+06   2.0000E+04   Tag Tires
```

TABLE 3: "CHECK" FILE OF TEST BUS MODEL, PRINTED BY A'GEM (CONT'D)

```
 20     .    .       .         .         .      0  1  0  0  0  0   8.7661E+05   2.0000E+04
 21     .    .       .         .         .      0  0  1  0  0  0   8.0310E+05   2.0000E+02
 22    11   13   -4.1020   -1.0710    -.5020    1  0  0  0  0  0   8.8565E+06   2.0000E+04   Tag Tires
 23     .    .       .         .         .      0  1  0  0  0  0   8.7661E+05   2.0000E+04
 24     .    .       .         .         .      0  0  1  0  0  0   8.0310E+05   2.0000E+02
 25     1    2    4.3430     .5910    -.7000    0  0  1  0  0  0   1.8000E+05   0.0000E+00   Front Air Springs
 26     1    2    4.3430    -.5910    -.7000    0  0  1  0  0  0   1.8000E+05   0.0000E+00   Front Air Springs
 27     1    5   -2.3750     .4640    -.5000    0  0  1  0  0  0   2.0000E+05   0.0000E+00   Drive Air Springs, Right Sid
 28     1    5   -3.2520     .4640    -.5000    0  0  1  0  0  0   2.0000E+05   0.0000E+00   Drive Air Springs, Right Sid
 29     1    5   -2.3750    -.4640    -.5000    0  0  1  0  0  0   2.0000E+05   0.0000E+00   Drive Air Springs, Left Side
 30     1    5   -3.2520    -.4640    -.5000    0  0  1  0  0  0   2.0000E+05   0.0000E+00   Drive Air Springs, Left Side
 31     8    5   -2.9320     .0810    -.8270    1  0  0  0  0  0   2.6270E+07   1.0000E+03   Bushings, A-Links
 32     .    .       .         .         .      0  1  0  0  0  0   4.3784E+06   5.0000E+02
 33     .    .       .         .         .      0  0  1  0  0  0   2.6270E+07   1.0000E+03
 34     .    .       .         .         .      0  0  0  1  0  0   2.3540E+04   1.0000E+01
 35     .    .       .         .         .      0  0  0  0  1  0   2.1340E+03   1.0000E+01
 36     .    .       .         .         .      0  0  0  0  0  1   2.3540E+04   1.0000E+01
 37     8    1   -3.5410     .5070    -.8560    1  0  0  0  0  0   2.6270E+07   1.0000E+03   Bushings, A-Links
 38     .    .       .         .         .      0  1  0  0  0  0   4.3784E+06   5.0000E+02
 39     .    .       .         .         .      0  0  1  0  0  0   2.6270E+07   1.0000E+03
 40     .    .       .         .         .      0  0  0  1  0  0   2.3540E+04   1.0000E+01
 41     .    .       .         .         .      0  0  0  0  1  0   2.1340E+03   1.0000E+01
 42     .    .       .         .         .      0  0  0  0  0  1   2.3540E+04   1.0000E+01
 43     9    5   -2.9320    -.0810    -.8270    1  0  0  0  0  0   2.6270E+07   1.0000E+03   Bushings, A-Links
 44     .    .       .         .         .      0  1  0  0  0  0   4.3784E+06   5.0000E+02
 45     .    .       .         .         .      0  0  1  0  0  0   2.6270E+07   1.0000E+03
 46     .    .       .         .         .      0  0  0  1  0  0   2.3540E+04   1.0000E+01
 47     .    .       .         .         .      0  0  0  0  1  0   2.1340E+03   1.0000E+01
 48     .    .       .         .         .      0  0  0  0  0  1   2.3540E+04   1.0000E+01
 49     9    1   -3.5410    -.5070    -.8560    1  0  0  0  0  0   2.6270E+07   1.0000E+03   Bushings, A-Links
 50     .    .       .         .         .      0  1  0  0  0  0   4.3784E+06   5.0000E+02
 51     .    .       .         .         .      0  0  1  0  0  0   2.6270E+07   1.0000E+03
 52     .    .       .         .         .      0  0  0  1  0  0   2.3540E+04   1.0000E+01
 53     .    .       .         .         .      0  0  0  0  1  0   2.1340E+03   1.0000E+01
 54     .    .       .         .         .      0  0  0  0  0  1   2.3540E+04   1.0000E+01
 55     1    5   -3.3160    0.0000    -.2930    0  0  0  1  0  0   2.7407E+04   0.0000E+00   Roll Bar
 56     1   10   -3.7410     .7150    -.3830    1  0  0  0  0  0   1.0000E+08   0.0000E+00   Tag Axle Pivots
 57     .    .       .         .         .      0  1  0  0  0  0   1.0000E+08   0.0000E+00
 58     .    .       .         .         .      0  0  1  0  0  0   1.0000E+08   0.0000E+00
 59     .    .       .         .         .      0  0  0  1  0  0   1.0000E+07   0.0000E+00
 60     .    .       .         .         .      0  0  0  0  0  1   1.0000E+07   0.0000E+00
 61     1   11   -3.7410    -.7150    -.3830    1  0  0  0  0  0   1.0000E+08   0.0000E+00   Tag Axle Pivots
 62     .    .       .         .         .      0  1  0  0  0  0   1.0000E+08   0.0000E+00
 63     .    .       .         .         .      0  0  1  0  0  0   1.0000E+08   0.0000E+00
 64     .    .       .         .         .      0  0  0  1  0  0   1.0000E+07   0.0000E+00
 65     .    .       .         .         .      0  0  0  0  0  1   1.0000E+07   0.0000E+00

   THERE ARE   9 LINEAR DIRECTED CONNECTION ELEMENTS

 Join  End1      ----POSITION----        End2     ----POSITION----         STIFFNESS    DAMPING
  No
  66     1    4.124   -.464   -.673     2    4.200    .654   -.606     6.5675E+06   1.0000E+04   Front Panhard Rod
  67     1    4.946    .370   -.660     2    4.343    .395   -.606     1.3135E+07   1.0000E+04   Front, Upper Trailing Link,
  68     1    4.946    .606   -.305     2    4.343    .632   -.251     1.3135E+07   1.0000E+04   Front, Lower Trailing Link,
  69     1    4.946   -.370   -.660     2    4.343   -.395   -.606     1.3135E+07   1.0000E+04   Front, Upper Trailing Link,
  70     1    4.946   -.606   -.305     2    4.343   -.632   -.251     1.3135E+07   1.0000E+04   Front, Lower Trailing Link,
  71     1   -3.541    .587   -.323     5   -2.932    .587   -.300     1.3135E+07   1.0000E+04   Drive Axle Leading Link, Rig
  72     1   -3.541   -.587   -.323     5   -2.932   -.587   -.300     1.3135E+07   1.0000E+04   Drive Axle Leading Link, Lef
  73     1   -3.886    .699   -.828    10   -4.000    .699   -.612     9.0000E+04   0.0000E+00   Tag Axle Air Spring, Right
  74     1   -3.886   -.699   -.828    11   -4.000   -.699   -.612     9.0000E+04   0.0000E+00   Tag Axle Air Spring, Left
```

TABLE 3: "CHECK" FILE OF TEST BUS MODEL, PRINTED BY A'GEM (CONT'D)

```
   THERE ARE   8 NONLINEAR CONNECTION ELEMENTS
Join  End1     ----POSITION----       End2    ----POSITION----        STIFFNESS     DAMPING
No
75     1    4.555    .676   -.990    2    4.555    .629   -.430    0.0000E+00   0.0000E+00    Front Shock Absorber, Right
                                                                      Vel          Force
                                                                  -8.0000E-02  -5.5500E+02
                                                                  -8.0000E-03  -1.7500E+02
                                                                  -5.0000E-03  -1.0000E+02
                                                                   5.0000E-03   1.0000E+02
                                                                   8.0000E-03   1.5000E+02
                                                                   2.4000E-02   3.0000E+02
                                                                   4.0000E-02   5.1500E+02
                                                                   8.0000E-02   1.5400E+03
76     1    4.100   -.673   -.990    2    4.100   -.597   -.430    0.0000E+00   0.0000E+00    Front Shock Absorber, Left
                                                                      Vel          Force
                                                                  -8.0000E-02  -5.5500E+02
                                                                  -8.0000E-03  -1.7500E+02
                                                                  -5.0000E-03  -1.0000E+02
                                                                   5.0000E-03   1.0000E+02
                                                                   8.0000E-03   1.5000E+02
                                                                   2.4000E-02   3.0000E+02
                                                                   4.0000E-02   5.1500E+02
                                                                   8.0000E-02   1.5400E+03
77     1   -2.564    .575   -.968    5   -2.564    .576   -.365    0.0000E+00   0.0000E+00    Drive Shock Absorber, Right
                                                                      Vel          Force
                                                                  -8.0000E-02  -2.2000E+02
                                                                  -4.0000E-02  -1.6200E+02
                                                                  -8.0000E-03  -1.3800E+02
                                                                  -5.0000E-03  -1.0000E+02
                                                                   5.0000E-03   1.0000E+02
                                                                   8.0000E-03   1.5000E+02
                                                                   2.4000E-02   3.0000E+02
                                                                   4.0000E-02   5.1500E+02
                                                                   8.0000E-02   1.5400E+03
78     1   -2.564   -.575   -.968    5   -2.564   -.575   -.365    0.0000E+00   0.0000E+00    Drive Shock Absorber, Left L
                                                                      Vel          Force
                                                                  -8.0000E-02  -2.2000E+02
                                                                  -4.0000E-02  -1.6200E+02
                                                                  -8.0000E-03  -1.3800E+02
                                                                  -5.0000E-03  -1.0000E+02
                                                                   5.0000E-03   1.0000E+02
                                                                   8.0000E-03   1.5000E+02
                                                                   2.4000E-02   3.0000E+02
                                                                   4.0000E-02   5.1500E+02
                                                                   8.0000E-02   1.5400E+03
79     1   -3.069    .575   -.968    5   -3.069    .575   -.366    0.0000E+00   0.0000E+00    Drive Shock Absorber, Right
                                                                      Vel          Force
                                                                  -8.0000E-02  -2.2000E+02
                                                                  -4.0000E-02  -1.6200E+02
                                                                  -8.0000E-03  -1.3800E+02
                                                                  -5.0000E-03  -1.0000E+02
                                                                   5.0000E-03   1.0000E+02
                                                                   8.0000E-03   1.5000E+02
                                                                   2.4000E-02   3.0000E+02
                                                                   4.0000E-02   5.1500E+02
                                                                   8.0000E-02   1.5400E+03
80     1   -3.069   -.575   -.968    5   -3.069   -.575   -.365    0.0000E+00   0.0000E+00    Drive Shock Absorber, Left T
                                                                      Vel          Force
                                                                  -8.0000E-02  -2.2000E+02
                                                                  -4.0000E-02  -1.6200E+02
                                                                  -8.0000E-03  -1.3800E+02
                                                                  -5.0000E-03  -1.0000E+02
                                                                   5.0000E-03   1.0000E+02
                                                                   8.0000E-03   1.5000E+02
                                                                   2.4000E-02   3.0000E+02
                                                                   4.0000E-02   5.1500E+02
```

TABLE 3: "CHECK" FILE OF TEST BUS MODEL, PRINTED BY A'GEM (CONT'D)

```
                                                              8.0000E-02   1.5400E+03
 81     1  -3.744    .781  -1.088    10   -4.208    .781   -.722    0.0000E+00   0.0000E+00    Tag Axle Shock Absorber, Rig
                                                                 Vel           Force
                                                             -8.0000E-02  -2.2000E+02
                                                             -6.4000E-02  -2.0000E+02
                                                             -4.0000E-02  -1.6200E+02
                                                             -2.4000E-02  -1.5600E+02
                                                             -8.0000E-03  -1.3800E+02
                                                             -5.0000E-03  -1.0000E+02
                                                              5.0000E-03   1.0000E+02
                                                              8.0000E-03   1.3800E+02
                                                              2.0000E-02   1.7500E+02
                                                              4.0000E-02   2.5000E+02
                                                              6.0000E-02   3.7000E+02
                                                              8.0000E-02   5.0000E+02
 82     1  -3.744   -.781  -1.088    11   -4.208   -.781   -.722    0.0000E+00   0.0000E+00    Tag Axle Shock Absorber, Lef
                                                                 Vel           Force
                                                             -8.0000E-02  -2.2000E+02
                                                             -6.4000E-02  -2.0000E+02
                                                             -4.0000E-02  -1.6200E+02
                                                             -2.4000E-02  -1.5600E+02
                                                             -8.0000E-03  -1.3800E+02
                                                             -5.0000E-03  -1.0000E+02
                                                              5.0000E-03   1.0000E+02
                                                              8.0000E-03   1.3800E+02
                                                              2.0000E-02   1.7500E+02
                                                              4.0000E-02   2.5000E+02
                                                              6.0000E-02   3.7000E+02
                                                              8.0000E-02   5.0000E+02

   There are  0 Constrained DOFs due to pins
CALCULATIONS COMPLETE -  10:43 AM  THU.,  4  JUN., 1987
```

2.2 MODELS USING SUBSTRUCTURES - SUBSTR

In suspension systems, there are often bodies or groups of bodies of small mass, which are not the primary focus of the suspension designer, but do serve to connect more fundamental elements of the system. An example would be the diagonal links connecting the drive axle and the carbody of the Test Bus Model. To be correct, each link must be modelled as a rigid body having several degrees-of-freedom. They cannot be modelled simply as diagonal springs since a lateral displacement of one end does not directly develop a force through the link (a force would be developed in a diagonal spring).

If the motions of the links themselves are not of primary interest, it is advantageous to determine their "equivalent" stiffness and damping as a connection element in the overall suspension. In this way, the number of degrees-of-freedom in the suspension system can be reduced, which in turn reduces the computation time and complexity in the simulation modules.

The method of substructures is provided to minimize the number of degrees-of-freedom in the main analysis. In a substructure, a group of bodies of small mass can be represented as its own small system. A right-handed co-ordinate system must be set up for the substructure with reference axis directions consistent with the main suspension model but with a co-ordinate origin chosen arbitrarily for convenience and independent from the remainder of the suspension system. An input description file is created for A'GEM exactly in the same manner as for any suspension model. However, two reference bodies, given zero mass and inertias and zero values for their centre of gravity location, must be included as the first two rigid body elements in the substructure system.

These two reference bodies will be assigned rigid body numbers and will represent the bodies at each end of the substructure to which the substructure is attached. That is, some bodies in the substructure will be connected to these reference bodies by connection elements. An example of a substructure for the Test Bus Model is shown in Figure 3, and the corresponding description file for the substructure which is input to A'GEM is given in Table 4.

The file output from A'GEM is then processed by program SUBSTR. This small program computes the equivalent stiffness and damping matrices for the substructure as it would be as a connection element in the main suspension model. The output file from SUBSTR may then be input into the simulation modules BUSRID and BUSHAN together with the main suspension model. The equivalent stiffness and damping matrices are incorporated into the main suspension system model by the simulation modules before performance calculations are carried out.

In the above example, 12 degrees-of-freedom are removed from the system by using the simple substructure. Performance calculations for the suspension system are identical, except that detailed information on the motions of the diagonal links as rigid bodies is not available.

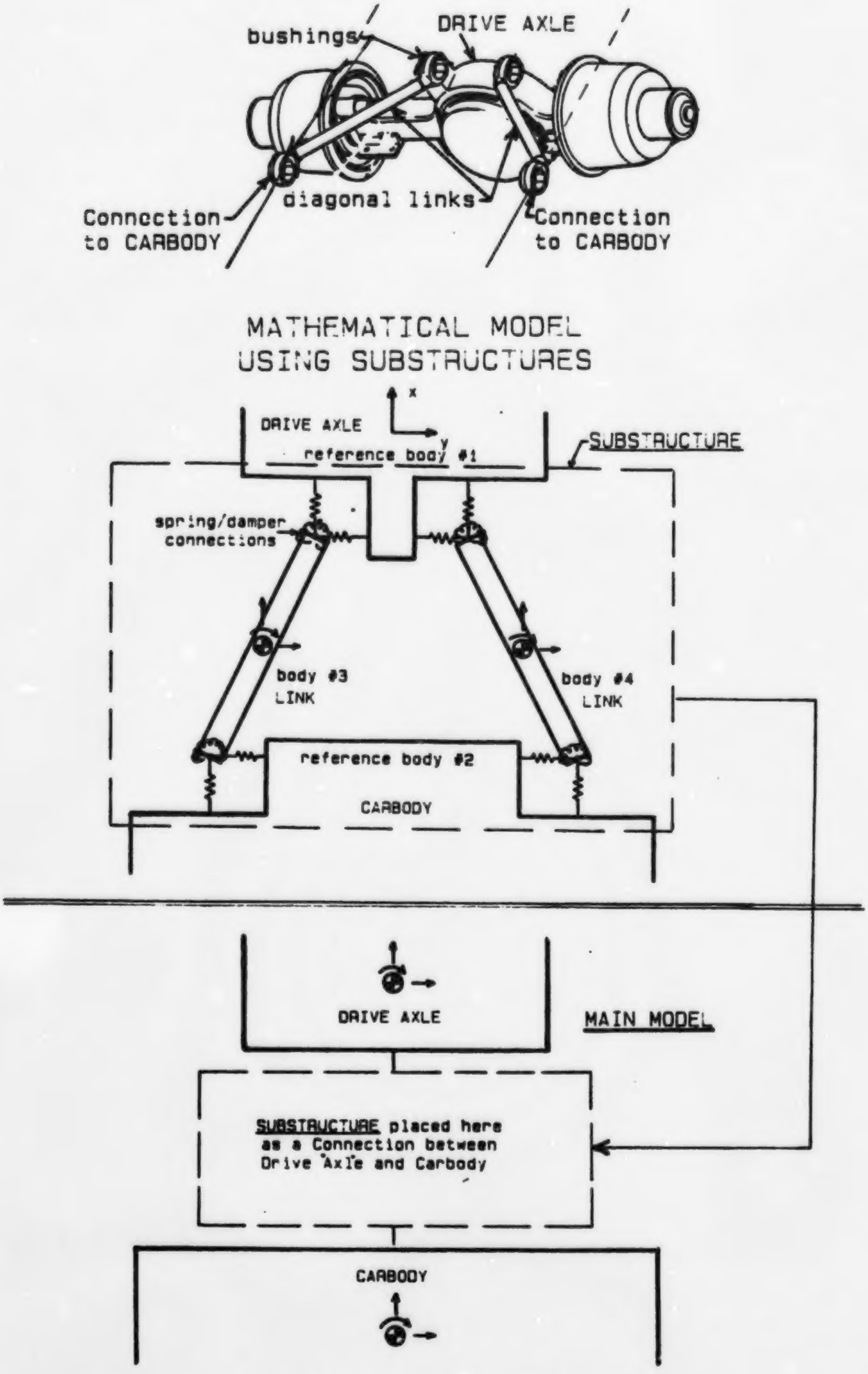

bushings
DRIVE AXLE
diagonal links
Connection to CARBODY
Connection to CARBODY
MATHEMATICAL MODEL USING SUBSTRUCTURES
x
y
DRIVE AXLE
reference body #1
SUBSTRUCTURE
spring/damper connections
body #3 LINK
body #4 LINK
reference body #2
CARBODY
DRIVE AXLE
MAIN MODEL
SUBSTRUCTURE placed here as a Connection between Drive Axle and Carbody
CARBODY

FIGURE 3 - EXAMPLE SUBSTRUCTURE FOR TEST BUS MODEL

File: /SUSPENSION/SUBA

```
 *
 *       A'GEM Input File
 *
 BUS MODEL SUBSTRUCTURE A     -----TRIANGULAR LINKS
 *
 *  Rigid Body Data
 *
     4
   0.0000       0.0000       0.0000     * Ref Body #1 - Drive Axle
 .0000E+00   .0000E+00   .0000E+00   .0000E+00   .0000E+00   .0000E+00   .0000E+00
   0.0000       0.0000       0.0000     * Ref Body #2 - Bus Body
 .0000E+00   .0000E+00   .0000E+00   .0000E+00   .0000E+00   .0000E+00   .0000E+00
   -3.2370       .3350       -.8420     * A-link, right
 25.40      1.1508      -1.3956     .0000E+00    1.7030     .0000E+00    2.970
   -3.2370      -.3350       -.8420     * A-link, left
 25.40      1.1508      +1.3956     .0000E+00    1.7030     .0000E+00    2.970
    1      2      3      4      5      6
    7      8      9     10     11     12
   13     14     15     16     17     18
   19     20     21     22     23     24
 *
 *  Point Element Data
 *
    1
 .26270E+08  .43784E+07  .26270E+08  23540.      2134.0      23540.
  1.00E3        5.00E2       1.00E3     10.          10.          10.
    4       * Bushings, A-Links
    1     3    -2.9320       .0810      -.8270
    2     3    -3.5410       .5870      -.8560
    1     4    -2.9320      -.0810      -.8270
    2     4    -3.5410      -.5870      -.8560
 *
 *  Linear Directed Element Data
 *
    0
 *
 *  Nonlinear Directed Element Data
 *
    0
 *
 *  Pinned Joint Data
 *
    0
```

TABLE 4 - EXAMPLE SUBSTRUCTURE SYSTEM DESCRIPTION FILE INPUT TO A'GEM

3.0 RIDE PERFORMANCE - BUSRID

The ride simulation examines the motions of the bus suspension system model in response to vertical forces of interaction at the tire/road interface.

The highway bus rides on tires which follow the road surface vertically as the vehicle moves over it. In order to simulate the dynamic motions, the tires (or the contact patches between the tires and the road) must be included as individual bodies in the vehicle model before it is input into the A'GEM module. Dual tires may be considered as a single tire, in which case, the standard 3-axle highway bus generally requires 6 tire bodies. Each of these bodies must further be given a vertical degree of freedom for the ride simulation.

In BUSRID, the vertical displacements of each tire in time are used as the input to the simulation. That is, each tire vertical degree-of-freedom is constrained to move in its specified manner in time, and the equations of motion are integrated to determine the time response of the other degrees-of-freedom in the vehicle model to these inputs. General inputs include:

- the vertical "shaking" of any specified tire or tires at given amplitude and phase in a sinusoidal manner;
- distinct vertical inputs in time to each of the front tires which are followed in time (dependent on the vehicle speed and tire location) by the same vertical inputs to the succeeding tires.

The stiffness, damping and mass matrices for the linear equations of motion are simply passed to the simulation from the A'GEM Module output file. The user specifies which rigid bodies of the model are the tires and their input displacements with time. Initially, there is one equation of motion for each degree of freedom in the system.

The simulation automatically rearranges the equations, separating out (partitioning) the tire vertical degrees of freedom in order from front to rear of the bus and, for equal longitudinal location, from right to left side. These equations are manipulated algebraically and need not be integrated since these degrees-of-freedom already have specified displacements with time. The tires are grounded laterally in the simulation (constrained to not move laterally) by removing any tire lateral degrees-of-freedom from the system of equations. This reduces the number of equations to be solved and ensures a purely vertical input. It should be noted that vertical and lateral springs representing the tire stiffness are still included in the simulation model; the tire "body" is just a representation of the wheel/road interface.

The remaining equations may then be integrated to determine the time histories of all the other degrees of freedom in the vehicle model relative to the ground.

In BUSRID, before the integration is carried out, the user is given the option to make a preliminary examination of the vehicle model. With this option, the rigid bodies of the system are statically placed under the force of 1 g acceleration (gravity) in the vertical direction. The tires are constrained vertically to their ground positions, and the linear stiffness equations are solved to determine the displacements of the vehicle degrees-of-freedom. The system is thus easily checked for obvious modelling errors, and the static normal force through each tire is also found.

In addition, the linear equations are also solved to determine the characteristic frequencies (eigenvalues) and modes (eigenvectors) of the linear system. This generally assists in the overall examination of the suspension system.

The time step integration procedure in BUSRID utilizes a RUNGE-KUTTA-FEHLBERG routine which carries out a fourth and fifth order integration simultaneously. Results are compared, and if a specified tolerance is met, integration proceeds to the next time step. If the tolerance is not met, smaller time steps are taken until the tolerance is satisfied or a specified minimum step has been reached, at which point the routine is stopped.

Control data input to program BUSRID is in the following order with variable names as they appear in the code:

1. 1 integer
   IOP

   IOP = option control flag.
   If IOP = 0, a time step integration is done
   If IOP = 1, a static load check (1 g. vertical acceleration) and an eigenvalue analysis is done
   If IOP = 2, both analyses are done

   Note: When a new model is being examined, it is helpful to use IOP = 1 initially to check for errors in the model parameters.

2. 20 alphanumeric characters (or less)
   FILIN

   FILIN = file name of the large binary file, output from A'GEM, which contains all matrices and information about the main suspension system model.

3. 1 integer
   NSUB

   NSUB = the number of substructure files used to complete the model

Note: If NSUB = 0, omit records 4 and 5. Otherwise, the following sequence of input data (records 4 and 5) must be repeated NSUB times.

4. 20 alphanumeric characters (or less)
   FILINS

   FILINS = file name of a substructure file, output from SUBSTR, containing equivalent stiffness and damping matrices for a substructure.

5. 2 integers
   NB1, NB2

   NB1 = the body number to which one end of the substructure is connected (the first reference body used in the creation of the substructure when input to A'GEM).

   NB2 = the body number to which the other end of the substructure is connected (the second reference body used in the creation of the substructure when input to A'GEM).

6. 1 integer, several integers up to a maximum of 20
   NWH, (Wheel (I), I = 1, NWH)

   NWH = the number tire bodies (wheels) used in the suspension model.

   Wheel (I) = the body numbers of each tire body

   Note: The body numbers were assigned in the model description input to A'GEM

Note: The remaining records are not required if IOP = 1 in Record 1.

7. 1 Integer
   IPROF

   IPROF = Control flag indicating the type of vertical profile to be input at the wheels.

   IPROF = 1: All wheels are driven over a half sine-wave bump, at a speed given in record 8.

IPROF = 4: only the right wheels are driven over a half sine-wave bump, at a speed given in record 8.

IPROF = 2: all wheels, except the front wheels, are subjected to a sine-wave input, all in phase.

IPROF = 3: all wheels, except the front wheels, are subjected to a sine-wave input, but out of phase side to side.

IPROF = 5: all wheels are subjected to a road profile defined in an input file named in record 10.

Note: IPROF = 2,3 were specific inputs used in conjunction with tests done to validate a particular suspension model.

Note: Record 8 is required only if IPROF = 1 or IPROF = 4.

8. 4 real numbers
VEL, AMP, LMBDA, BSTRT

VEL = forward velocity of the bus (m/s)

AMF = amplitude (height) of the bump (m)

LMBDA = wavelength of the bump (m)

BSTRT = the starting location of the bump relative to the front wheel of the bus (m)

Note: Record 9 is required only if IPROF = 2 or IPROF = 3

9. 2 reals
FREQ, AMP

FREQ = frequency of the sine-wave input to the wheels (Hz)

AMP = amplitude of the sine-wave input to the wheels (m)

Note: Records 10 and 11 are required only if IPROF = 5

10. 20 alphanumeric characters (or less)
FILINP

FILINP = file name of a binary file containing a road profile over which the bus will be driven.

11. 1 real
VEL

VEL = forward velocity of the bus (m/s)

12. 6 reals (time-step integration parameters)
    TO, TEND, DT, HMIN, HMAX, TOLA

TO = starting simulation time for the integration (s)

TEND = end simulation time for the integration (s)

DT = time step at which all parameters are to be written to output files for examination (s)

HMIN = minimum time step (s) to be used in the actual integration scheme. The integrator will automatically adjust the time step in order to meet the required accuracy. If the necessary time step for accuracy becomes less than HMIN, the program will terminate.

HMAX = the maximum time step (s) to be used in the actual integration scheme. The integrator will automatically adjust the time step in order to meet the required accuracy. However, the time step will be limited to a maximum of HMAX.

TOLA = the absolute tolerance allowed in the accuracy of each integrated value in the system.

Note: Records 13 and 14 are optional

13. 20 alphanumeric characters (or less)
    FILIND

FILIND = file name of binary file containing the initial conditions of displacement for every degree-of-freedom in the system.

14. 20 alphanumeric characters (or less)
    FILINV

FILINV = file name of a binary file containing the initial conditions of velocity for every degree-of-freedom in the system.

A sample control file for BUSRID for the Test Bus Model is given in Table 5.

```
Line No                      File:    /SUSPENSION/BU1EIN

   1        0
   2       BU1ENLI01.GEM
   3         0
   4         6      3  4    6  7    12  13
   5        4
   6       20.00    -0.025400   4.0000     0.50
   7          0.0      3.0    0.005     1.E-6    0.010    2.E-3
```

4.0 HANDLING PERFORMANCE - BUSHAN

The handling simulation module examines the motions of the bus suspension system model in response to lateral forces of interaction between the tire and the road. In order to simulate the dynamic motions and forces, the tires (or the contact patches between the tires and the road) must be included as individual bodies in the vehicle model before it is input into the A'GEM module. Dual tires may be considered as a single tire. Each of these tire bodies is generally given lateral and longitudinal degrees-of-freedom in the model in order to accommodate tire inputs for the handling simulation. This general approach to including tire bodies in the model means that any number of tires can be simulated. Buses of any design fit easily into the structure of the simulation program.

The angle of the steered road wheels (tires) relative to the ground are used as the principal inputs to the handling simulation. The front steer angles are generally under driver control and are those usually specified. However, the capability is included to specify the wheel angles on any of the tires as a function of time.

In BUSHAN, the carbody (the largest mass) is taken as a reference body and is assigned the task of containing the basic degrees-of-freedom for the simulation. The motion of this body is completely described in terms of the forward velocity, lateral velocity and yaw rate. The assumption is made that the vehicle is travelling over smooth level pavement and that vertical motions will be small. The acceleration of every other component of the bus model is found by applying a relative motion analysis.

Tire forces arise from a misalignment between the direction a tire is headed and the direction it is moving (slip angle). The total velocity vector of the tire is therefore not aligned with the tire and a lateral force of interaction between the tire and the roadway is developed.

BUSHAN allows the user to specify particular tire force characteristics or to make use of a cubic variation of lateral force with slip angle which includes the saturation force levels. Since the slip angles are functions of the velocity vectors of the tire, the tire velocities are computed at every time step. Since the tire forces also depend on the normal load through the tire, this normal force is also computed at each time step.

The time step integration procedure in BUSHAN is the same as that used in BUSRID.

Control data input to program BUSHAN is in the following order with variable names as they appear in the code:

1. 20 alphanumeric characters (or less)
   FILIN

   FILIN = file name of the large binary file, output from A'GEM, which contains all matrices and information about the main suspension system model.

2. 1 integer
   NSUB

   NSUB = the number of substructure files used to complete the model.

   Note: the number of substructure files used to complete the model.

Note: If NSUB = 0, omit records 3 and 4. Otherwise, the following sequence of input data (records 3 and 4) must be repeated NSUB times.

3. 20 alphanumeric characters (or less)
   FILINS

   FILINS = file name of a substructure file, output from SUBSTR, containing equivalent stiffness and damping matrices for a substructure.

4. 2 integers
   NB1, NB2

   NB1 = the body number to which one end of the substructure is connected (the first reference body used in the creation of the substructure when input to A'GEM).

   NB2 = the body number to which the other end of the substructure is connected (the second reference body used in the creation of the substructure when input to A'GEM).

5. 2 integers
   NTTYPE, NTR

   NTTYPE = number designating the type of tire used on the bus

   NTTYPE = 0: Use an empirical cubic variation of lateral tire force versus slip angle.

   NTTYPE = 1: Michelin 12R22.5 radial tire

   NTTYPE = 2: Michelin Pilote 11/80R22.5 low profile tire

NTR = number of tires (wheels) used in the suspension model

Note: Curves of lateral force versus slip angle for various normal forces have been included in the computer code for the 2 tires indicated above. As more data becomes available for other tires, allowance has been made for the inclusion of additional data in the code.

6. NTR integers
(NTRNO (I),I=1,NTR)

NTRNO (I) = the body numbers of each tire in the model. These body numbers were assigned in the model description input to A'GEM.

7. NTR integers
(NAXNO (I),I=1,NTR)

NAXNO (I) = the body numbers of the axle corresponding to each tire given in record 6. The axle body numbers were assigned in the model description input to A'GEM.

Note: the axle body number is used in defining the steered road wheel angle for each tire. The steered wheel angle is defined as the relative angle between the tire and the axle if an axle body number is given. If the axle body number corresponding to a given tire is made zero, then the steered wheel angle is taken as the relative angle between the tire and the bus body.

8. NTR real numbers
(FTNORM (I),I=1,NTR)

FTNORM (I) = the static normal force (N) through each of the tires given in record 6.

Note for record 8:

(a) These normal forces are output from program BUSRID when a static load check is requested (IOP = 1).

(b) If a single tire body represents dual tires, a negative sign should be placed in front of the corresponding force to signal the simulation. Correct calculations of slip force will then be made for dual tires.

9. 1 real number
VEL

VEL = forward velocity of the bus (m/s)

10. 1 integer
ISTYPE

ISTYPE = control flag indicating the type of steering input to the tires

ISTYPE = 0: A step steer or steady state steer will be applied to the steered tires.

ISTYPE = 1: A step steer will be applied to the steered tires, and the step will be smoothed by a cubic smoothing function

ISTYPE = 2: A steering input file will be read in, containing the wheel angles of each steered tire as a function of time.

Note: Records 11 and 12 are required only if ISTYPE = 0,1.

11. 1 integer, 2 real numbers, 1 integer
NSTRS, TSO, DELTS, NSPTS

NSTRS = the number of tires to be steered

TSO = the starting time (s) at which the steering angles are defined in record 12

DELTS = the step size in time (s) at which the steering angles are defined in record 12

NSPTS = the total number of times at which the steering angles are defined in record 12

12. 1 integer, NSPTS real numbers
IT, (STRDAT (IR,J),J=1,NSPTS)

IT = the body number of the tire whose steering is being defined

STRDAT (IR,J) = the steering angles of the tire, defined at steps in time defined in record 11

Notes for record 12:

(a) the steering angles are defined at times given as:
t = TSO + (J-1) * DELTS

(b) record 12 will be repeated NSTRS times

(c) All tires not included in records 12 will be given zero steered angle in the program.

Note: Record 13 is required only if ISTYPE = 2

13. 20 alphanumeric characters (or less)
FILINS

FILINS = the file name of a binary file containing the wheel angles of each steered tire as a function of time. The first column of this file will be time, and succeeding columns will be the steered wheel angles of each wheel at the corresponding time.

Note: At each time in the time step integration, the steered wheel angles will be interpolated from the table entered in this file.

14. 6 reals (time-step integration parameters)
TO, TEND, DT, HMIN, HMAX, TOLA

TO = starting simulation time for the integration (s)

TEND = end simulation time for the integration (s)

DT = time step at which all parameters are to be (stored) written to output files for examination

HMIN = minimum time step (s) to be used in the actual integration scheme. The integrator will automatically adjust the time step in order to meet the required accuracy. If the necessary time step for accuracy becomes less than HMIN, the program will terminate.

HMAX = the maximum time step (s) to be used in the actual integration scheme. The integrator will automatically adjust the time step in order to meet the required accuracy. However, the time step will be limited to a maximum of HMAX.

TOLA = the absolute tolerance allowed in the accuracy of each integrated value in the system.

Note: Records 15 and 16 are optional.

15. 20 alphanumeric characters (or less)
FILIND

FILIND = file name of a binary file containing the initial conditions of displacement for every degree-of-freedom in the system.

16. 20 alphanumeric characters (or less)
FILINV

FILINV = file name of a binary file containing the initial conditions of velocity for every degree-of-freedom in the system.

A sample control file for BUSHAN for the Test Bus Model is given in Table 6.

```
                          File:    /SUSPENSION/BU1EINHAN
Line No
   1      BU1ENL101.GEM
   2        0
   3        1      6
   4         3   4   6   7   12   13
   5         2   2   5   5   10   11
   6       28663.   28415.   -42892.   -41782.   13748. 12505.
   7        14.00
   8        0
   9       2    0.0  0.25   6
  10         3    0.0   0.0   3.5   7.0   10.5   10.5
  11         4    0.0   0.0   3.5   7.0   10.5   10.5
  12      0.0    0.5    0.01     1.E-8    1.E-2   .005
```

TABLE 6: Control File Input to BUSHAN